

SHIPBUILDING
THEORETICAL AND PRACTICAL

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SHIPBUILDING,

THEORETICAL AND PRACTICAL.

Illustrated by a Series of Engravings,

FROM DRAWINGS FURNISHED BY SOME OF THE MOST EMINENT BRITISH SHIPBUILDERS.

BY

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P R E F A C E.

THIS Treatise provides a complete system of information on the Art of Shipbuilding, and on the scientific principles on which it is founded, at a price within the means of the general body of practical men who are engaged in that Art.

There is a growing interest felt in the education of British Naval Architects, and a strong desire that it should not fall short of what is now being accomplished in France; evinced, amongst other facts, by the continually increasing attendance at the Royal School of Naval Architecture, and by the commencement of the special study of the scientific principles of Shipbuilding at Universities. Hence one object of this Work is, to lay down those scientific principles in as plain and clear a manner as possible, for the benefit especially of young students who may desire to be well grounded, in order that they may afterwards advance without hesitation in the prosecution of their honourable and useful profession.

For the convenience of practical men, all rules for ordinary use are stated in words at length, and illustrated by numerical examples worked out in detail. At the same time, principles are expressed in algebraical symbols as well as in words, where the use of symbols is conducive to clearness. Those portions of the text of the First Division in which algebraical symbols occur, are inclosed in brackets, thus []: throughout the rest of the book, algebraical symbols are used in the foot-notes only.

The Treatise consists of Seven Divisions.

The First Division, entitled "Hydraulics of Shipbuilding," explains the scientific principles which guide the Naval Architect in designing a ship, so that she shall possess the properties required of her, as to displacement, steadiness, and speed, in order that she may fulfil her practical object; and in computing the power which will be required to drive her at her intended speed, whether by sails or steam. The calculations required in those processes are illustrated in full detail, by examples taken from actual ships whose practical success has proved the goodness of their design. Some instances of faulty design are given also, and the nature of their defects explained. So far as this Division relates to buoyancy and stability, the principles which it states have long been well known; nevertheless, some improvements have recently been made in the art of conveniently applying them to practice, and these are explained and exemplified. Much of the theory of the oscillations of ships has hitherto been published in scientific journals and transactions only; and such is the case also with the whole theory of the frictional resistance of vessels as modified by their form, which is illustrated by examples from actual ships ranging in size from the *Warrior* to the *Fairy*. Mr. Scott Russell's "wave-theory" as to the dimensions and figures of ships, and its application to practice, are explained; but of course, for full details on the subject of that theory, reference must be made to the writings of its Author.

The Second Division, entitled "Geometry of Shipbuilding," describes the methods by which the model and plans of an intended ship are constructed, and the figure and dimensions of her parts laid off.

The Third Division, entitled "Strength of Materials as applied to Shipbuilding," sets forth the facts and principles known as to the strength of the materials of which ships are built, whether timber or iron, and the application of those facts and principles to practice. It contains the results of the latest and best investigations, theoretical and practical, on that subject; and is accompanied by very extensive tables of the strength, elasticity, and heaviness of iron, steel, timber, and other substances.

Annexed to the Third Division is an Appendix, containing a summary of the Rules adopted by Societies of Underwriters, together with their Tables of Scantlings and of Durability, illustrated by Plates; and for the permission to reprint those Rules, Tables, and Plates, the Authors of this Treatise have to return their best thanks to the Committee of Lloyd's, and to the Committee of the Liverpool Registry.

The Fourth Division, entitled "Practical Shipbuilding," commences with an account of the special properties of the materials used in Shipbuilding, of the signs of good or bad quality in those materials, and of their durability, preservation, and treatment. It then describes the processes gone through in shaping and putting together those materials, and explains the structure, building, and fitting up of a ship in detail, and the process of launching her.

The Fifth Division, entitled "Masts, Sails, and Rigging," treats of the adaptation of those parts to the ship's power of bearing them, and of their dimensions, proportions, materials, and structure.

The Sixth Division, entitled "Marine Steam Engineering," sets forth the scientific principles of the propulsion of a ship by steam-power, and the practical rules which regulate the construction of her engines. The investigation, based altogether on the results of successful practice, from which are deduced the rules for adapting propelling instruments, whether jets, paddles, or screws, to the vessels that they are to drive, has hitherto appeared in the Transactions of the Institution of Naval Architects only. The rules for the power, efficiency, and dimensions of engines, and for their expenditure of heat, water, and fuel, are wholly based on and in conformity with the dynamical theory of heat, and are at the same time reduced to a degree of simplicity which it is believed has not hitherto been attained. They are accompanied by the requisite tables, and by a pair of engraved diagrams; one showing the relations between the pressure, volume, and temperature of steam, and the other, the proportion borne by the mean pressure to the initial pressure at different grades of expansion.

The Seventh Division relates to "Shipbuilding for purposes of War;" and owing to the changing condition of that art, and the deficiency of practical data on which principles can be founded, that Division has more of the nature of a brief summary of existing information than of a detailed system of rules. In everything that relates to buoyancy, stability, strength, speed, propelling power, and handiness, the fundamental principles of shipbuilding for purposes of war are the same with those of shipbuilding for other purposes, and are explained in the first Six Divisions of the Treatise; leaving to be considered in the Seventh Division, the subject of Armour for War-Ships; its materials, its structure, and its power of resisting the explosive energy of gunpowder as conveyed by projectiles; and also its various arrangements, in the form of belts, broadside-armour, and turrets. The broadside system of armament is illustrated by eight detailed plates of H.M.S. *Warrior*—a vessel whose design in point of form, proportions, and arrangement is acknowledged to be one that can scarcely be surpassed, and whose armour has been proved to be perfectly efficient against those projectiles and charges of powder which it was intended to resist. The turret system, undoubtedly the best yet invented for carrying a few very heavy guns, is illustrated by a design prepared by Mr. C. F. Henwood, according to the plans of Captain Cowper P. Coles, R.N.

In conclusion, the Authors have to express their deep sense of obligation to the Lords Commissioners of the Admiralty, for having authorized the use in this Work of copies from the original plans of ships of the Royal Navy, most carefully prepared expressly for this Work by Mr. W. Hinde; and they have also to return their best thanks, for the prompt and liberal manner in which they have been supplied with drawings and information, to many eminent Naval Architects, Shipbuilders, and Engineers, whose names appear on the Plates, or are mentioned in the course of the Book.

May, 1866.

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E R R A T A.

Page 71, 2nd column, 1st line, *for* "roof" *read* "root."

Page 82, 2nd column, 7th line from the bottom, *for* "third" *read* "sixth."

Page 141, 2nd column, at the end of the formula of Example IX. in the table, after "3 C," insert "— 12 A B C."

Page 165, 2nd column, 12th line, *for* "400" *read* "320."

" " 20th " " "400" " "320."

" " 22nd " " "100" " "80."

" " 23rd " " "500" " "400."

" " 25th " " "10,000" " "8000."

" " 26th " " "1250" " "1000."

Page 170, 2nd column, 26th line, *for* "NEW" *read* "IRON."

Page 199, 1st column, 19th line, *for* "positive" *read* "negative."

" " 21st " " "negative" " "positive."

Page 220, 1st column, in the Table headed "EXAMPLE," opposite the words "Height, CH, of base of sail above centre of lateral resistance," *for* "24 feet" *read* "22 feet"

Page 238, 2nd column, last line of text, *for* "35" *read* "55."

Page 251, 2nd column, 16th line from bottom, *for* "8250" *read* "5250."

Page 252, 2nd column, 8th line from bottom, *for* "one-fifth" *read* "one-sixth."

" " 3rd, 4th, and 5th lines from bottom, the passage from the word "Divide" to the word "thickness" should be as follows:—

"Divide the ratio in which the depth of an arm is to be greater than its thickness by twice the total number of arms in the wheel."

Page 253, 1st column, 3rd line, *for* "breadth" *read* "depth."

" " 4th line should be as follows:—

$$\sqrt[3]{\left(\frac{4}{2 \times 84}\right)} = \sqrt[3]{0.0238} = 0.288.$$

Page 283, 1st column, 1st line, *before* "the distance, B C" *insert* "half."

Also add the following at the end of the paragraph:—"This construction, as well as that in the foot-note relating to double slides, is demonstrated by Mr. J. M'Farlane Gray in his 'Geometry of the Slide-Valve.'"

In the Table of the Properties of Iron and other Metals, between pages 150 and 151, column of Specific Gravities, opposite IRON, WROUGHT, *for* "7" *read* "7.7."

DIVISION FIRST.

HYDRAULICS OF SHIPBUILDING.

CHAPTER I.

NATURE AND OBJECTS OF THE HYDRAULICS OF SHIPBUILDING.

ARTICLE 1. *The Qualities sought in a Ship* depend mainly on the fact, that the ship, with her burden, has to be safely and steadily carried by, and propelled through, the water, her movements being at all times under control. Setting aside, then, for the present, the qualities of strength and durability, which belong to a later division of this treatise, the qualities of a ship which conduce to her efficient support by and propulsion through the water may be thus summed up:—

BUOYANCY, to enable her to carry her burden without either sinking too deep in the water, or floating too lightly on it:—

STABILITY, that she may tend to “right herself” when disturbed from an upright position, and may never, under the action of winds, waves, or other disturbing causes, deviate further from that position than is consistent with convenience and safety; and also, that her movements may neither be so extensive nor so abrupt as to strain or damage her structure or contents:—

SPEED sufficient for her purpose, with due regard to economy in the means whereby such speed is obtained:—

The quality of WORKING WELL, which it would be difficult to find any single word to express. In a vessel propelled by steam alone, it consists chiefly in ready and quick answering to the helm; in a sailing vessel, it embraces also “weatherliness,” and the performing of various manœuvres with promptness and certainty.

All those qualities depend mainly on forces exerted between the ship and the fluids by which it is surrounded—viz., the water and the air; and therefore the means of obtaining them depend to a great extent on principles belonging to the sciences of hydrostatics, or the balance of fluids, and hydrodynamics, or the motion of fluids. The practical application of those branches of science is commonly known by the term “hydraulics:” whence the title of the “Hydraulics of Shipbuilding” has been given to the present division of this treatise.

Besides knowing how to obtain separately each of the qualities that are sought in a ship, it is necessary that the naval architect should know how to combine those qualities in the manner best suited for the use to which the ship is to be applied, and how to insure that the means adopted for obtaining one of them shall not be injurious to the others. This is the kind of knowledge whose application constitutes DESIGN in naval architecture.

The present chapter will be devoted to a general account of the principles to be observed in designing a ship, leaving the details of their application to be explained further on.

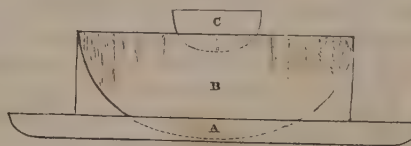
2. The *Buoyancy* of a ship depends on the following principles:—

I. That in order that a body may remain steady in a given position, the forces acting upon it must be balanced; which, in the case of there being two forces, means that they must be equal, and directly opposed, to each other:—

II. That a body plunged into a still fluid is urged downwards by its own weight, and pressed upwards by the fluid with a force equal and opposite to the weight of the volume of fluid which the body displaces:—

III. That consequently, in order that a given body, such as a ship, may float steadily in a given position in smooth water, the weight of the volume of water displaced must be equal to the weight of the body; and the total upward pres-

Fig. 1.



sure of the water, which is equal and opposite to the weight of the water displaced, must be directly opposed to the weight of the body.

The quantity of water displaced by a ship is called her *displacement*; and may be expressed either by its volume (for example, as so many cubic feet), or by its weight (for example, as so many tons, a ton being the weight of 35 cubic feet of ordinary sea-water, or 35·9 cubic feet of fresh water).

The principle No. II. above stated is easily demonstrated by the following reasoning. The total pressure exerted on the solid body by the neighbouring particles of fluid is the same with that which was previously exerted on the mass of fluid whose place the solid body occupies; and that mass of fluid was *in equilibrio*; therefore the total pressure exerted on it was equal and directly opposed to its weight.

That the weight of the water displaced by a floating body is equal to that of the body and all its contents, may be experimentally proved by apparatus within the reach of every one. Take two vessels, A and B, as represented in Fig. 1; place one within the other, and fill the upper one with water to the brim: then take another empty vessel, C, and lower it gradually into the water in B, until it is supported by the pressure of the water. When C is at rest, a volume of water equal to that displaced by it has run over into A; and if this water be placed in one side of a pair of scales, and the vessel C in the other, they will be found to balance each other. Replace C in the water, and gently drop some heavy material, such as sand or shot into it, and more water will overflow; remove C with the material it contains carefully to one end of the balance, and add to the water before put in that which was caused to overflow by the introduction of the material into C, and, as before, the water will balance the vessel and its contents. This may be often repeated, until C sinks nearly to its upper part, and it will be found in every experiment that the weight of the water which has overflowed from B is always equal to that of C, and of the material it contains, provided great care be taken that none of the water is lost, and that none adheres to the outside of the vessels.

From these practical proofs of the equality which always exists between the weight of the floating body with its contents and that of the water displaced, we also learn that for every weight put on board of a ship there is an equal weight of water displaced by it. Of the knowledge of that fact we shall avail ourselves when we have to discuss the question of stowage.

In consequence of the necessity for this equality of the weight of the ship and of the water displaced, we perceive that, if the water in which a ship floats at different times differs in density, there will be a corresponding difference in the immersion of the ship. Now, the weight of a cubic foot of sea water is a little more than 64 lbs., and the weight of a cubic foot of river water about one-fortieth less. A line-of-battle ship like the *Agamemnon*, when ready for sea, weighs about 5000 tons, and it requires 26 tons to increase her immersion one inch; consequently, were such a ship to come direct from sea into the river, keeping precisely the same weights on board, she would sink $\frac{5000}{26} \times \frac{1}{64}$ inches, or 5 inches (very nearly) deeper in the river than when at sea. Ships which have flat floors, and are full forward and aft—approximating, in fact, to the form of a rectangular box—as are some colliers, barges, &c., will sink just one fortieth deeper when in river water, than when at sea.

We have already proved that a floating body presses downwards, and is pressed upwards, by forces equal to its weight. If, now, a light air-tight vessel, such as an empty cask, be placed in the water, it will float, until it is loaded with a weight equal to the difference between its weight and that of an equal volume of water. Let one end of a rope be secured to the weight, and the other end to the air-tight vessel; throw the weight into the water, and as soon as the equilibrium has been restored, it will be found that the vessel is nearly at the surface of the water, and the weight hanging vertically under it, supported by the tension of the rope. It would be entirely immersed, were it not for the effect which the immersion of the weight and the rope has on the vessel, that effect being to raise it a little out of the water—the part emersed being equal to the volume of the weight and of the rope, diminished by a quantity of fluid the weight of which is the same as that of the rope.

On this principle, sunken vessels are often recovered in the following manner:—At low water, a number of empty casks, air-tight caissons, or one or two ships or barges, are attached by strong ropes or hawsers to parts of the sunken ship, and the ropes are hove in tight. As the tide rises, the vessels become more and more immersed in the water until the weight of the additional volume of water displaced by the whole of them equals the force necessary to raise the ship. When tide is nearly at its height, the vessels, with the sunken ship under them, are removed towards the shore until she touches the ground again. If the ship be then in such a position that the falling tide will leave her above water when at its lowest, the vessels are cast off: but if not, they are hove down as before, and the process described is repeated.

The number of air-tight vessels necessary to raise a sunken ship may be thus approximated to. On the sunken ship the pressure downwards is the weight of the ship and of the cargo, and the pressure upwards is the weight of a volume of water equal to that occupied by the materials of the ship and by the cargo. If the ship be built of wood, the specific gravity of the mass does not much exceed unity—that is, the weight of the whole mass would be about the same as that of an equal volume of water. There would then remain to be overcome by the water-tight vessels a pressure equal to the weight of the cargo when placed in water. This pressure can often be found very readily. When known, we must have a number of water-tight vessels, such that their weight, together with the weight of cargo when in water, shall equal the weight of the volume of water displaced by these vessels. If the ship be built of iron, with the usual amount of wood work, the weight of the whole is about five times the weight of a volume of water equal to the bulk of the materials. In addition, therefore, to the difference between the weight of the cargo and that of a volume of water equal to it—that is, to the weight of the cargo in water—four-fifths of the entire weight of the ship has to be overcome by the pressure of the immersed water-tight vessels.

To find the weight of a vessel by computing the weight of the several parts composing it would be a problem almost impracticable because of its complexity. But after the ship is completed and floating in the water, since we know that her weight is equal to that of the water she displaces, we have only to find the cubic contents of the part immersed, multiply it by the weight of a unit of volume of water, and the product is the weight of the ship and everything in it.

When the ship is of any regular form, the volume of displacement may be found by certain mathematical rules. Ships are, however, usually of no regular figure, their sections conforming to no other law than the will of the constructor; consequently, whatsoever methods may be employed to find the displacement, they cannot be more than approximations. Those approximations, however, as we shall hereafter show, may be made to approach the exact displacement as nearly as we please.

In determining how much the displacement of a proposed ship ought to be, the naval architect will consider, in the first place, what burden the vessel is to carry, whether in the shape of cargo stores, armament, or otherwise; and then, knowing from experience what proportion the weight of a ship of the kind intended, with her equipments, bears to her lading, he will calculate that weight, and thence the whole displacement required. In steam-vessels provision must be made in the total lading for the weight of engines and fuel, which will depend on the dimensions and form

of the vessel, the intended speed, the length of voyage, and the construction and economy of the engine.

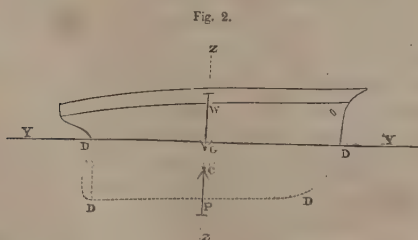
Particulars of the proportionate weights of ships and their lading will be given further on in the treatise; in the meanwhile it may be stated, that the weight of a ship with her equipments (engines *not* included) usually ranges from about *one-third* to *one-half* of the whole displacement; iron vessels in general approaching the lighter of those limits, and wooden vessels the heavier.

3. *Centres of Gravity and of Buoyancy.*—In the previous article have been considered the consequences of the principle, that the weight of the ship and the pressure of the water must be *equal*. We have now to consider the consequences of the principle, that they must be *directly opposed*; which, as both forces act vertically, the weight downwards, and the pressure upwards, means that they must act *in the same vertical line*.

A force acting in a single line, or, what is the same thing, at a single point, is purely imaginary, being an idea introduced in order to make mechanical calculations possible. The action of every force is diffused throughout some space; for example, the action of a body's weight is diffused throughout the whole bulk of the body, and the action of a pressure, over the surface of contact of two bodies or parts of a body which press each other. Nevertheless, it is always possible to find a single point in a body acted upon by gravity or by pressure, such that the effect of the weight or pressure on the body as a whole is the same as if its action were really concentrated at that point. That point, in the case of a body's weight, is called the *centre of gravity* of the body; in the case of a pressure, it is called a *centre of pressure*; and when the pressure is that of a fluid in which a solid body floats, the *centre of buoyancy*. The single or concentrated force which is thus conceived to be equivalent to a given diffused force, is called the *resultant* of the diffused force.

It is a necessary consequence of the principle No. II. stated at the beginning of the preceding article, that the centre of buoyancy of a floating body must occupy the place of the centre of gravity of the mass of fluid displaced; and hence that point is sometimes called the *centre of displacement*.

Let Fig. 2 represent a ship, floating in smooth water, of which Y Y is the surface. The weight of the whole ship acts as if it were



concentrated at the centre of gravity of the ship, G, and may be represented by the arrow, W, pointing vertically downwards. The immersed part of the ship, D D D D, displaces a volume of water, whose weight is equal to that of the ship. Let C be the centre of gravity of that volume of water, that is, the centre of buoyancy; then the pressure of the water against the ship acts as if it were a single force, P, equal to the ship's weight, acting vertically upwards through the centre of buoyancy, C; and as the weight and pressure must be directly opposed, the centres of gravity, G, and of buoyancy, C, must be in the same vertical line, Z Z.

It may here be explained, that the particles of water press against the ship's bottom horizontally and obliquely, as well as vertically; the direction of the pressure exerted by each particle being at right angles to that part of the ship's bottom which it touches; but the horizontal parts of the whole pressure exactly balance each other; so that the resultant acts vertically upwards, as already stated.

The precise position of the centre of gravity of a ship is always to a certain extent capable of adjustment; that is, the ship's "*trim*" may be altered after she is afloat, by suitable stowage of her lading; but a skilful naval architect will always be able to fix on the "*trim*" beforehand, in designing a ship, to such a degree of approximation, that the subsequent adjustment shall not cause any inconvenience.

4. *Stability in Smooth Water.*—A body which is free to move, is said to be *stable*, if, when disturbed from its position of balance or steadiness, it tends to *right itself*, or return to that position. If, on the other hand, it tends to deviate further from that position, or upset, it is said to be *unstable*.

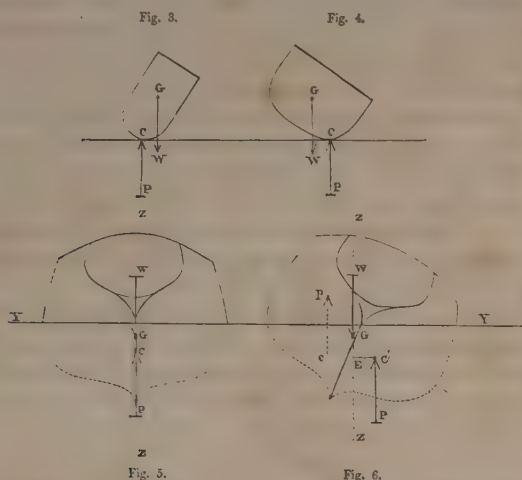
A ship is always stable as regards vertical disturbances, that is, rising above or dipping below her position of steady floating; for when she rises out of the water, her displacement is diminished, and there is an excess of the weight over the supporting pressure, tending to bring her down again; and when she dips deeper into the water, her displacement is increased, and there is an excess of the supporting pressure over the weight, tending to make her rise again.

The kind of disturbance of a ship's position which it is of primary importance to consider, is that which consists in *heeling*, or leaning over to one side. Similar disturbances in a longitudinal plane, known as *pitching* and *scending*, have also to be considered, although they are of less importance than heeling. All these are disturbances of angular position, and stability against them all depends on similar principles; so that it will be sufficient for the purposes of the present chapter to explain on what stability against heeling, or transverse stability, depends.

It has already been stated, that in order that a pair of forces applied to one body may balance each other, they must not only be equal in amount and opposite in direction, but *directly* opposed to each other—that is, they must act in opposite directions along the *same* straight line. When a pair of equal forces act in opposite directions along *parallel*, but not identical lines, they no longer balance each other, but constitute what is called a *couple*, tending to turn the body into a new angular position. When the angular position of such a body as a ship is disturbed, the weight and the supporting pressure, which originally were a pair of directly opposed equal forces, producing balance, become a couple; and the body is stable or unstable, according as that couple is a *righting couple* or an *upsetting couple*.

It may facilitate the understanding of this subject to give an illustration taken from the mechanics of solid bodies. Figs. 3 and 4 represent two blocks, each with a rounded base, resting on a level platform. Either of those blocks may be balanced on its rounded end, by so placing it that the upward pressure of the platform, exerted against the point of support, may act in a line passing through the centre of gravity of the block. If the block with the sharper curvature at the base, Fig. 3, is disturbed, the weight, W, acting through the centre of gravity, G, and the pressure exerted by the platform at the point of support, C, form an *upsetting*

couple, which makes the block fall over on its side. If the block with the flatter base, Fig. 4, is disturbed, the weight acting through G , and the pressure of the platform acting through C , form a right-



ing couple, which makes the block return to its position of balance. The condition of a ship as regards stability, is analogous to that of the latter block.

Fig. 5 represents an end view of a ship floating upright in smooth water, of which YY is the surface. G is the ship's centre of gravity; C the centre of buoyancy, in the same vertical line ZZ ; W represents the weight of the ship, exactly balanced by the equal and opposite resultant pressure, P . Fig. 6 represents the same ship, having heeled over through a certain angle towards the right. The weight of the ship, W , continues to act through the same centre of gravity, G , in the same vertical line, ZZ ; but in consequence of the new form assumed by the immersed part of the ship, or displacement, the centre of buoyancy shifts into a new position, C' ; and in a properly designed ship, that new position lies to the same side of the vertical line, ZZ , as that towards which the ship has heeled; so that the weight, W , and the resultant pressure, P , form a *righting couple*, tending to bring the ship back to the upright position. Had the new centre of buoyancy, through a faulty design, lain to the other side of ZZ , as at c , the weight and pressure would have formed an *upsetting couple*.

The *moment* of a couple is the name given to the magnitude of its tendency to turn the body on which it acts, and is computed by multiplying either of the two equal forces of which the couple consists, by the perpendicular distance between the parallel lines of action of the forces, which distance is called the *arm* or *leverage* of the couple. The moment of the righting couple which acts on a ship at some fixed angle of heel, is called her *moment of stability* at that inclination. For example, in Fig. 6, the moment of stability is the weight of the ship, w , multiplied by the horizontal distance of the new centre of buoyancy, C' , from the vertical line, ZZ , traversing the ship's centre of gravity; that is—

$$W \times EC'.$$

The moment of stability required for different sorts of vessels has been ascertained by practical experience. Examples will be given later in the treatise. For the present it may be stated by way of illustration, that a common value for the moment of stability in large vessels at an angle of heel of fifteen degrees, is the weight of the ship acting with a leverage of one foot. The

power of a vessel to carry sail obviously depends mainly on her stability.

5. *Steadiness in Rough Water.* Although a certain amount of stability in a ship is absolutely necessary, an excess of that quality becomes an evil, for the following reasons:—

I. A ship of great stability is quick in her rolling motion; and if the stability be excessive, the rolling may be so quick as to strain and damage her structure and contents.

II. The same form and proportions which make a ship very stable in smooth water, tend also to make her accompany the waves in their motions. This to a certain extent is necessary, but if it goes too far, causes inconvenience and danger.

III. It is dangerous for a ship in rolling to *keep time* with the waves, because in that case each successive wave increases the extent of the ship's rolling; and the best way to avoid that danger is to take care that the ship shall roll more slowly than the waves. The time of a ship's rolling is affected by the distribution of the weight of the ship and lading, as well as by the stability. To distinguish the tendency of a ship to keep *upright to the surface of the water*, whether level or sloping, from the tendency to keep *truly upright* in rough water, the former may be called *stiffness*, and the latter *steadiness*.

6. *Easy Rolling* is insured by avoiding excessive quickness of rolling, as already mentioned; and also by so designing the form of the ship's hull, that when she heels over, the pressure of the water shall tend to make her simply roll back again, and shall not tend at the same time to make her pitch, scend, or rise and fall bodily.

7. *Speed and Resistance.*—The resistance opposed by the water to the progress of a ship depends on the speed with which the ship moves through the water, and on the figure and dimensions of the vessel, and the smoothness of her immersed surface. So far as the resistance depends upon speed, it is well ascertained by experience that for a given vessel, and within the limits of speed to which that vessel is suited, the resistance is sensibly proportional to the square of the speed, being fourfold for a double speed, ninefold for a triple speed, and so on; in other words, it is proportional to the height from which a body must fall to acquire the velocity of the vessel. When the speed of the vessel, however, is urged beyond certain limits depending on her dimensions and figure, the resistance begins to increase sensibly faster than the square of the speed; the reason being, that the wave or swell raised in front of the vessel begins to have a sensible effect in adding to the extent of surface acted upon by the resistance.

Our knowledge of the manner in which the resistance is affected by the dimensions and figure of the vessel is still imperfect, although it has of late made much progress. During the last century attempts were made to deduce a theory of the resistance of ships from that of the impulse of jets of water against flat surfaces; but the results arrived at were so utterly inconsistent with those of practical experience that the theory has long ago been abandoned as useless; and indeed this was to have been expected, from the want of resemblance between the circumstances of the two cases compared together. Better success has attended the theory which considers the resistance as being analogous to that met with by a stream in flowing along a channel; that is to say, as depending on a certain degree of viscosity, or stiffness, in the water; and by means of that theory, marine engineers have of late years been enabled in various instances to compute beforehand the engine-power

required to drive an intended vessel at a given speed with accuracy sufficient for practical purposes. Examples of such calculations will be given further on. The viscosity of the water acts in two ways; by producing a direct backward drag, exerted by the particles of water on the skin of the vessel; and by causing a heaping up of water against the bow of the vessel as compared with the stern.

Independently of any knowledge of the particular mode of action of the water in resisting the progress of a vessel, it is obvious, and must always have been obvious to common observation, that the vessel which makes the least commotion in the water is the least resisted. Thus it has been known from remote antiquity that the length of a ship should be greater than her breadth; and that fine ends, both at the bow and stern, causing the particles of water to be displaced and replaced gradually, are favourable to speed.

It has been left for correct theoretical views to show conclusively, what had already been only partially and imperfectly known through practical experience, that there are limits beyond which great length as compared with breadth ceases to be an advantage, and becomes a cause of increased resistance.

In the proportion of the length of a ship to her breadth, the ratio of 4:1 was seldom exceeded until after the introduction of steam navigation. Then gradually increasing proportions were introduced, with continually improving results, until the ratio of 7:1 was reached; and it was naturally taken generally for granted that an unlimited increase of length and slenderness would cause an unlimited diminution of resistance. This, however, was not found to be the case in practice, which concurs with theory to show that the proportion of 7:1, or thereabouts, is very nearly the utmost that, is attended with advantage in vessels whose draught of water is not specially limited; and that greater proportions, such as 8:1, 9:1, or 10:1, are advantageous under special conditions only, such as limited draught of water, or limited breadth of channel.

Another principle which is known independently of the precise mode of action of the particles of water is this—that the resistances of vessels of *similar figures*, but different dimensions, at the same speed, are nearly proportional to their *surfaces*—that is, to the squares of their linear dimensions, or to the squares of the cube roots of their displacements; so that, for example, if there be two ships of precisely similar figures, one of 1000 tons displacement, and the other of 1331 tons, which are to each other as the cube of 10 to the cube of 11; then the resistances of those vessels at the same speed will be to each other nearly as the square of 10 to the square of 11; that is, as 100 to 121. This principle, however, ceases to be exact in extreme cases; so that it is not applicable, for instance, to the comparison of real ships with small models. To make experiments on the resistance of models available for arriving at correct conclusions respecting vessels on the large scale, the velocities of the models must be kept within certain limits, to be afterwards specified.

It is evident that the smoothness or roughness of a vessel's bottom must materially affect her resistance. The bottom of every ship, how smooth soever it may have been originally, tends to become crusted in time with shells and weeds, which increase the adhesion of the water. It is very common to find the resistance increased by about a fourth from this cause; and occasionally it is increased more. The means of preventing or removing such incrustation consists mainly in coating the vessel with some substance which shall from time to time scale off in thin flakes, carrying the incrustation with it, and leaving the bottom clean.

Such is the action of the copper sheathing of wooden ships, and of various paints and other compositions used for protecting iron ships.

8. *Fairness—Models.*—Although some knowledge has been gained of the effect of using certain definite curves, such as the curve of versed-sines or harmonic curve, the trochoid or rolling-wave curve, and some others, for the water-lines or horizontal sections of ships; and although the effect of those and various other forms on the motion of the water has been theoretically investigated, it cannot yet be said that any particular form has been conclusively demonstrated to be the form of least resistance for a given displacement. Still, it has always been admitted by all naval architects, that the figure of a ship should be what mathematicians call “*continuous*,” and shipbuilders “*FAIR*.” A *fair line* is a line in which there is not only no sudden change of direction, but no sudden change of curvature, and a *fair surface* is one whose sections are all fair lines. The fairness of the water-lines, or horizontal sections, is of the highest importance in the form of a ship; next in importance is the fairness of the vertical longitudinal sections, called *bow* and *buttock lines*; and sometimes, to test the fairness of the intended form of a ship still further, oblique sections, called *diagonal* or *riband lines*, are drawn. The naval architect judges of the fairness of lines by the eye; and sometimes, if a model of the vessel is before him, by the sense of touch. To show the form of lines distinctly, models are made of layers of differently coloured wood.

9. *Propulsion by machinery*, although of more modern invention (if we except rowing and paddling by hand) than propulsion by sails, is much simpler in its principles, and will therefore be considered first. The fundamental principle of the action of every propeller is the same, whether it is an oar, a paddle, a screw, a jet, or any other contrivance. The propeller drives backward a certain quantity of water at a certain speed; in so doing, it presses backwards against the water with a force depending on the quantity of water driven back, and the speed impressed upon it. The water presses forward against the propeller with an equal force; and that force (which is called the “*re-action*” of the water), being transmitted to the framework of the machinery and thence to the vessel, is what drives her forward. When the ship is starting from a state of rest, or increasing her speed, the driving force must be greater than the resistance; but so long as the speed is uniform, the driving force and the resistance are equal.

Some propellers act directly backwards on the water, or nearly so, like the feathering paddle; some, like the common paddle, act more or less obliquely in a vertical plane; some, like the screw, act obliquely in various directions. Due regard must be given to those circumstances; and also to that of the working of the propeller in water which has already been disturbed by the vessel.

The *engine-power* required to drive a given vessel at a given speed, depends on a mechanical principle which governs the action of all machines whatsoever—the *equality of the energy exerted to the work performed*. The *useful* part of the work performed in driving a vessel during a given time (such as a second), is found by multiplying the resistance by the distance through which the vessel is driven. To this has to be added the *wasteful work*, one part of which is performed in the following manner:—The propeller exerts backwards a force equal to that with which the vessel is driven forwards; but it exerts that force through a greater distance, viz.:—the *sum* of the distances through which the vessel is driven forward, and the water backward (the latter distance is

called the *ship*); so that the total work performed by the propeller is greater than the useful work, in the proportion in which the total backward speed of the propeller exceeds the forward speed of the ship—a proportion which ranges from $1\frac{1}{10} : 1$ to $2 : 1$. Some additional work is wasted in giving lateral and vertical motions to the water, and in overcoming the friction of the machinery. The sum of all those quantities of work, useful and wasteful, which are performed in a given time, is equal to the energy which must be exerted in the same time by the engine. The ratio which the useful work bears to the energy exerted, is called the *efficiency* of the propeller and its machinery; in some of the best examples of economy of power in steam-vessels, it is about $3 : 5$; in less economical examples it is sometimes as low as $2 : 5$, and perhaps even less.

When resistances are expressed in pounds, and are multiplied by the distances through which they are overcome, in feet, the products, or quantities of work, are said to be expressed in “foot-pounds.” If the total number of foot-pounds of energy exerted per second in driving a vessel be divided by 550, the quotient is the *real* or *indicated horse-power* of her engine.

Upon the power of the engine, and its system of construction and working, depends its weight (including that of its boilers), for which, as well as for the store of fuel, provision must be made in the displacement of the vessel, and to which regard must be had in considering questions of stability.

The details of the construction and working of marine steam-engines will form the subject of a special division of this treatise; but the principles of the action of propellers, of the adjustment of their dimensions to the speed of the vessel, and of the power required to work them, belong to the general subject of the hydraulics of shipbuilding.

10. *Propulsion by Sails* depends upon more complex principles than propulsion by machinery; because the direction of the wind, which supplies the propelling force, is seldom the same with that of the vessel's course, and often makes a great angle with it. The pressure of the air on the sails, moreover, does not depend on the *real* direction and velocity of the wind relatively to the ocean, but on its *apparent* direction and velocity relatively to the moving ship, and also on the position of the sails themselves.

The details of masts, sails, and rigging, belong to a later division of this book; but the general principles of the action of sails belong to the hydraulics of shipbuilding.

The pressure of the wind, diffused over the surface of the sails, is capable, like the pressure of water and the force of gravity (see Art. 3), of having its action on the ship as a whole represented by one resultant, traversing a point called the *Centre of Effort*, which is at a height above the deck depending on the figure, dimensions, and positions of the sails.

When that resultant is oblique to the ship's course, it is resolved, according to well-known principles, into two *component forces*—the longitudinal component, acting forwards parallel to the keel, which is the effective effort that drives the ship forward against the resistance of the water; and the transverse component, acting at right angles to the keel, which drives the ship to leeward; so that her real course or direction of motion, instead of being parallel to the keel, makes a small angle to leeward of that line, called the *angle of leeway*.

The angle of leeway depends on the proportion borne by the velocity with which the ship drifts to leeward, to her forward velocity; and the velocity with which she drifts to leeward is such that the resistance to her *transverse* motion through the water is exactly equal to the transverse component of the pressure of the wind on the sails. Those two forces, though equal and opposite, are not directly opposed; for the resistance acts below water, and the transverse pressure of the wind high above water; they therefore form a *couple* (Art. 4), tending to make the vessel heel over; and the vessel does heel over to leeward until the moment of the righting couple, or moment of stability, is sufficient to balance the heeling moment. Thus the power of a vessel to carry sail depends on her stability, and must be kept in view in designing her.

It is evident that, for the purpose of propulsion by sails, it is not sufficient that the hull of the vessel should be of such a shape as to meet with little resistance to forward motion through the water; its shape should also be such as to meet with great resistance to transverse motion or leeway.

11. The *Working or Manœuvring Qualities* of a ship depend on the combined action of the rudder, the propelling apparatus, the sails, and the figure of the ship's hull. In vessels under steam alone, the working qualities chiefly required are those of going readily astern as well as ahead, and of turning quickly and accurately, and in as small a circle as possible, under the action of the rudder. Sometimes, as in vessels driven by two screws with independent engines, the propelling apparatus may be used so as to turn the ship when required.

In vessels under sail, a much more complex combination of qualities is required. One of the most important is that of working well to windward, for which purpose it is essential that the ship should make little leeway, and should carry a *weather helm*; that is to say, that the action of the sails, unaided by that of the helm, should tend of itself to bring the ship's head towards the point from which the wind blows; this property, when in excess, is called *ardency*. It depends in a great measure on the position of the centre of effort of the sails, relatively to the resultant of the resistance to leeway. The manœuvring of a ship under sail, too, depends not merely on the action of the rudder and of the ship's hull on the water, but on the positions of the masts, and the figure and dimensions of the sails.

The working qualities of a ship are materially affected by her *trim*, or the position in which she floats. This depends on the position of her centre of gravity relatively to her centre of buoyancy; and the position of the centre of gravity depends partly on the stowage of the lading, as has been already stated in Article 3.

12. *Design*.—From the brief summary which has been given of the qualities sought in a ship, and the means of obtaining them, it is evident that every one of those qualities is more or less affected by every circumstance in the figure and dimensions of the ship, the distribution of her weight, and the nature and arrangement of her means of propulsion; and, consequently, that the naval architect must keep the whole of those qualities, and the whole of those circumstances, before his mind at once, in designing a ship. This subject will be treated of more fully in the last chapter of the present division.

CHAPTER II.

SUMMARY OF RULES IN MENSURATION AND MECHANICS.

ARTICLE 13. *Object of this Chapter.*—From the explanations given in the preceding chapter, it is evident that in designing a ship, the constructor has continual occasion to compute the areas of surfaces, and the volumes of solid figures, to find the resultants and centres of weights and pressures, and to determine the effects of forces, whether in producing balance or motion. The object of the present chapter is to give in a compact form those fundamental rules in mensuration and mechanics, to which reference will have to be made in the ensuing chapters of this division.

SECTION I.—MENSURATION OF AREAS AND VOLUMES.

14. *Units of Measure.*—In Britain, the linear dimensions of ships are stated in *feet*. The most convenient mode of subdividing the foot for all purposes of mensuration is that into decimal fractions; and that mode of subdivision will be employed throughout this treatise, except where there are special reasons for preferring the inch.

The areas of surfaces of ships are stated in *square feet*, and their volumes in *cubic feet*; which, like lineal feet, are subdivided into decimal fractions.

A peculiar unit of volume used in shipbuilding is the *ton*, which has different values according to circumstances. When the volume of water displaced by a ship is expressed in tons, the term *ton* means the volume of so much water as weighs a ton, or 2240 lbs. avoirdupois. That volume is—

For ordinary sea-water, about 35 cubic feet.
For fresh water, about 35·9 cubic feet.

For the water of estuaries and inland seas which receive large rivers (such as the Baltic Sea) it has intermediate values. When no other value is specified, it is to be understood that a *ton of displacement* means 35 cubic feet.

When the internal capacity of a vessel, above water as well as below, is expressed in tons, the word *ton* means from 94 to 100 cubic feet, according as the old or the present measurement is employed.

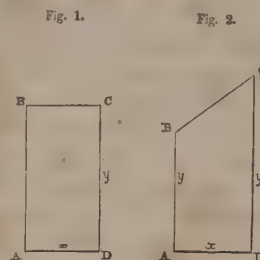
In the *metrical system of measures*, used in France and in many other continental countries, the unit of lineal measure in naval architecture is the *metre*, of area the *square metre*, and of volume the *cubic metre*; and the *ton* of water displaced is identical with the cubic metre. These units of measure bear proportions to the British units, which, together with their logarithms, are given in the following table:—

COMPARISON OF BRITISH AND METRICAL MEASURES.

	Ratios.	Logarithms.	Logarithms.	Ratios.
Feet in a metre,	3·2808992	0·5169929	1·4840071	0·30479449 metre in a foot.
Inches in a metre,	39·37079	1·5951741	2·4048259	0·02539954 metre in an inch.
Sq. feet in a sq. metre,	10·7643	1·0319858	2·9680142	0·0928997 square metre in a sq. foot.
Cubic feet in a cub. metre,	35·3166	1·5479787	2·4520218	0·0283163 cubic metre in a cubic foot.
Tons of 35 cubic feet in a cubic metre,	1·009046	0·0089107	1·9960893	0·9910855 } cubic metre in a ton of 35 cubic feet.

15. *The Area of a Rectangle* (such as ABCD, Fig. 1) is computed by multiplying together the lengths of its sides, AD and AB. In common language

the longer of those dimensions is called the length, and the shorter the breadth; but for mathematical purposes either dimension may be called the length, and the other the breadth, according to the circumstances of the case; or one side may be called the base, and the other the height.



[In algebraical symbols, if x denotes the length or the base, and y the breadth or the height, of a rectangle, the area is denoted by xy . The same calculation gives the area of an oblique-angled parallelogram, if y be taken to denote the length of one side, and x , instead of the length of a side, be taken to denote the perpendicular distance between the pair of sides whose length is y .]

16. *A Trapezoid* is a four-sided, straight-lined plane figure, of which two sides only are parallel. In Fig. 2, the trapezoid, ABCD, is of such a kind that its two parallel sides, AB and CD, are both at right angles to its base AD: the fourth side, BC, being oblique to the others; and the area of this figure is found by taking half the sum of the two breadths or parallel sides, AB and DC, and multiplying it by the length or base, AD.

The half-sum of the two parallel sides is the *mean breadth* of this figure; being the breadth of a rectangle of the same length and area with the trapezoid.

[In algebraical symbols, let the two parallel sides or breadths be denoted by y and y' , and the length or base by x , then the area is

$$\frac{y + y'}{2} x.$$

The same calculation gives the area of a trapezoid in which AD as well as BC is oblique to the two parallel sides, provided x be taken to denote the perpendicular distance between those sides.]

16A. *Abscissæ and Ordinates.*—It is here desirable to explain the use of these words in connection with the measurement of figures. The *base-line* along which the length of a figure is measured is sometimes called an *axis of abscissæ*, and any distance measured along that line is an *abscissa*. For example, in each of the Figures 1 and 2, the line AD is an axis of abscissæ, and its length (denoted by x in the algebraical formulæ) is an abscissa. Also, in each of the Figs. 3 and 4, further on, the line AD is an axis of abscissæ, and the distances, AE, AD, AG, of certain points in it from a fixed point, A, are abscissæ.

The distance of any point not in the base-line from that line is called an *ordinate*. For instance, in each of the Figs. 1 and 2, AB and DC are the ordinates of the points B and C respectively. In Fig. 1, those ordinates are equal to each other; in Fig. 2 they are unequal. Also, in Figs. 3 and 4, AB, EF, DC, GH, are respectively the ordinates of the points B, F, C, H, in the further boundaries of the figures.

[In algebraical symbols, it is usual to denote abscissæ by x , and ordinates by y ; distinguishing marks, such as accents or figures, being often employed when several different and unequal abscissæ or ordinates have to be referred to. For example, in Fig. 2, the two unequal ordinates are denoted by y and y' ; in Fig. 3 there are three ordinates, denoted by y , y' , y'' ; in some figures further on, it will be seen that a series of successive ordinates are denoted by affixing small figures *below* the letter y ; thus— y_0, y_1, y_2, y_3 , &c. These figures are affixed below, and not above; because if affixed above they might be taken for exponents of powers, instead of mere distinguishing marks.]

If a plane figure is bounded altogether by straight lines, its form and dimensions are completely known when the abscissæ and ordinates of its angles are known. In order that the form and dimensions of a curved boundary may be expressed exactly by means of abscissæ and ordinates, it is necessary that an algebraical equation should be known, expressing the relation which any ordinate whatsoever bears to the corresponding abscissa. But in many, and perhaps in most of the cases which occur in naval architecture, the curved lines under consideration are of figures for which no such general equation is known, or for which, if known, it is too complex for ordinary use; and then the figure of a curved boundary is expressed approximately with sufficient nearness to absolute accuracy for practical purposes, when the abscissæ and ordinates of a series of points in it are given; the nearness of the approximation being the greater, the closer and more numerous those points are. For example, in Fig. 5 (further on), the curve, BC, may be one for which no general algebraical equation between an abscissa and the corresponding ordinate is known; but still the figure of the curve may be expressed so as to enable it to be drawn, and various calculations to be made respecting it, with sufficient accuracy for practical purposes, if the particular ordinates corresponding to a series of particular abscissæ are known.

So far as plane figures are concerned, the rules to which the present section chiefly relates are those for calculating the area of such a figure from a given series of abscissæ and ordinates.

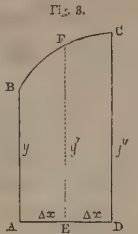
[In algebraical symbols, a plane curved line is expressed exactly when an equation of the general form,

$$y = \phi(x),$$

is given; meaning that the ordinate is a quantity depending in a given way on the abscissa. When the curve is expressed approximately by means of a series of points, a series of abscissæ are given, which may be denoted by 0, x_1, x_2, x_3 , together with the corresponding ordinates, y_0, y_1, y_2, y_3 , &c.; y_0 being taken to denote the ordinate at the "origin" or starting-point of the abscissæ. In most of the examples which follow, the ordinates are *equidistant*; so that the abscissæ increase by an *uniform interval or difference*, which is denoted by Δx ; in such cases the series of abscissæ is 0, $\Delta x, 2\Delta x, 3\Delta x$, &c. The area of a curve is denoted by the symbol $\int y dx$.]

17. *Areas of Parabolic Figures.*—In a restricted sense the word "parabola" is applied to the common parabola, being one of the conic sections; but in an extended sense, a figure is said to be parabolic when its boundary is a curve of such a kind that the value of any ordinate can be expressed by means of one or more terms, each of which is proportional to some power of the abscissa corresponding to that ordinate. The parabolic curve is said to be of the second order, the third order, &c., according to the exponent of the highest power of the abscissa. According to this definition, a parabola of the first order is a straight line. The common parabola is of the second order. The following are the rules for computing the areas of those parabolic figures which are of most frequent use in connection with naval architecture:—

I. *COMMON PARABOLA.*—Let ABCD, in Fig. 3, represent a plane figure, bounded by the base AD, two ordinates, AB, DC, at right angles to the base, and a parabolic curve of the second order, BFC. Bisect AD in E so as to divide the base into two equal intervals, and draw the middle ordinate, EF. Then, to find the area—*Add together the two endmost ordinates, and four times the middle ordinate; multiply the sum by one-third of the common interval of the ordinates (in this case one-half of the base); the product will be the area required.* (This is called "*Simpson's First Rule.*")



[In algebraical symbols, let y, y', y'' be the ordinates, Δx their common interval, then—

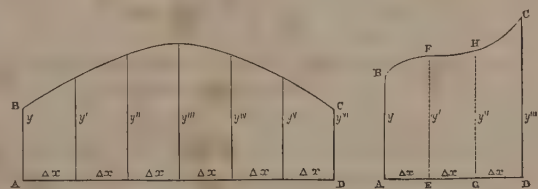
$$\int y dx = \frac{\Delta x (y + 4y' + y'')}{3}.]$$

Any figure in which three equidistant ordinates only are given, may be treated as approximately a parabolic figure of the second order.

II. *PARABOLA OF THE THIRD ORDER.*—Let ABCD, in Fig. 4, represent a plane figure, bounded by a base, AD, two ordinates,

Fig. 5.

Fig. 4.



AB, DC, at right angles to the base, and a parabolic curve of the third order, BFHC. Divide AD by the points E and G into three equal intervals, and draw the two intermediate ordinates, EF, GH. Then, to find the area—*Add together the two endmost ordinates, and three times the two intermediate ordinates; multiply the sum by three-eighths of the common interval (in this case one-third of the base); the product will be the area required.* (This is called "*Simpson's Second Rule.*")

[In algebraical symbols, let y, y', y'', y''' be the ordinates, Δx their common interval; then

$$\int y dx = \frac{3\Delta x (y + 3y' + 3y'' + y''')}{8}.]$$

Any figure in which four equidistant ordinates only are given, may be treated as approximately a parabolic figure of the third order.

Corresponding Rules for Parabolic figures of higher orders are given by various mathematicians; but no additional accuracy is obtained by them to compensate for their complexity.

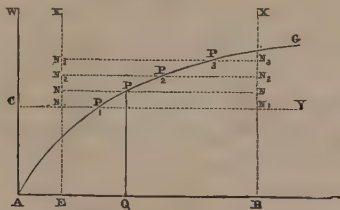
18. *Demonstrations of Rule for Parabolic Areas.*—The most complete and satisfactory demonstration of the applicability of Rule I. of the last article to the mensuration of all sorts of plane figures, and of the degree of accuracy of its results in different cases, is that given by Legendre in the appendix to the second volume of his "Traité des Fonctions Elliptiques;" but the methods employed in it are of a kind which would be out of place in the present treatise. When the precise degree of accuracy of the rule is not in question, it can be demonstrated in a more elementary way. The first of the following demonstrations is taken from a paper by Mr. F. K. Barnes, published in the "Mechanics' Magazine" for 1859: it supposes in the reader a knowledge of elementary algebra. The second supposes a knowledge of elementary geometry and arithmetic only. The third is even more elementary than the second, but not quite so conclusive; it is taken from Poncelet's "Mécanique Industrielle," and very slightly modified with a view to greater simplicity.

FIRST DEMONSTRATION.

The common form of the equation to the parabola is $y^2 = lx$, in which the origin of co-ordinates is at the vertex A; and l , which is constant in the same parabola, is called its parameter.

Let P be any point in the parabola, then, Fig. 6, $PQ^2 = l \times AQ$.

Fig. 6.



Instead of taking AW, AB as co-ordinate axes, let the dotted lines N1X and N1Y be taken, their intersection N1 being the new origin.

Let PN = Y, N1N = X, N1C = a, N1B = b.

Then PN = Y = BQ = AQ - AB = x - a;

or $x = Y + a$.

And N1N = X = PQ - N1B = y - b;

$\therefore y = b + X$.

Substituting these values for x and y in the common equation, we have $(b + X)^2 = l(Y + a)$. Multiplying out, dividing and transposing,

$$Y = \left(\frac{b^2}{l} - a\right) + \frac{2b}{l}X + \frac{X^2}{l},$$

which may be put in the form—

$$Y = A + BX + CX^2,$$

which is the general equation to the common parabola, whether the point N1 be within or without the curve; that is, whether the curve be concave or convex to the axis of abscissae N1X.

Let a_1 , a_2 , and a_3 be the lengths of the ordinates, corresponding to the distances 0, m , and $2m$ from N1, where m denotes N1N2 = N2N3, the interval between the ordinates.

$$a_1 = A \quad (1),$$

$$a_2 = A + Bm + Cm^2 \quad (2),$$

$$a_3 = A + 2Bm + 4Cm^2 \quad (3).$$

From these equations we find that

$$A = a_1,$$

$$B = \frac{4a_2 - a_3 - 3a_1}{2m},$$

$$C = \frac{a_3 - 2a_2 + a_1}{2m^2},$$

and the equation to the parabolic curve, passing through P1, P2, and P3, where P1N1 and N1N3 are the axes of Y and X respectively, is

$$Y = a_1 + \frac{4a_2 - a_3 - 3a_1}{2m}X + \frac{a_3 - 2a_2 + a_1}{2m^2}X^2.$$

Whence the ordinate at any point in N1N3 may be obtained.

Let N1N3 be divided into a very large number ($2n$) of equal parts, each part being equal to $\frac{m}{n}$; and, at the points of division

imagine ordinates $y_0, y_1, y_2, y_3, \&c., \dots y_{2n}$ drawn perpendicular to N1N3; and their extremities joined by straight lines, thus forming $2n$ small trapeziums. Now it is evident that by making n large enough, the very short lines, joining the extremities of the ordinates, may be made to coincide, as nearly as we please, with the curve; and the sum of the small trapeziums thus formed, will be as close an approximation as we wish to the parabolic curvilinear area required. From the general equation to the parabola.

$$y_0 = A;$$

$$y_1 = A + B \frac{m}{n} + C \frac{m^2}{n^2};$$

$$y_2 = A + B \frac{2m}{n} + C \frac{2^2 m^2}{n^2};$$

$$y^4 = A + B \frac{3m}{n} + C \frac{3^2 m^2}{n^2};$$

$$\&c. = \&c.$$

$$y_{2n} = A + B \frac{2n-1}{n}m + C \frac{(2n-1)^2 m^2}{n^2}.$$

$$\text{Area of first trapezium} = \frac{y_0 + y_1}{2} \times \frac{m}{n} = A \frac{m}{n} + B \frac{m^2}{2n^2} + C \frac{m^3}{2n^3};$$

$$\text{" second " } = \frac{y_1 + y_2}{2} \times \frac{m}{n} = A \frac{m}{n} + B \frac{3m^2}{2n^2} + C \frac{(1^2 + 2^2)m^3}{2n^3};$$

$$\text{" third " } = \frac{y_2 + y_3}{2} \times \frac{m}{n} = A \frac{m}{n} + B \frac{5m^2}{2n^2} + C \frac{(2^2 + 3^2)m^3}{2n^3};$$

$$\&c. = \&c.$$

$$\text{" last, or } 2nth = \frac{y_{2n-1} + y_{2n}}{2} \times \frac{m}{n} = A \frac{m}{n} + B \frac{m^2}{2n^2} + C \frac{\{2n-1\}^2 + 2n^2\} m^3}{2n^3}.$$

adding on both sides of the equations we have:—Sum of all the trapeziums =

$$\begin{aligned} & A \left\{ \frac{m}{n} + \frac{m}{n} + \frac{m}{n} + \&c. \text{ to } 2n \text{ terms} \right\} \\ & + B \frac{m^2}{2n^2} \left\{ 1 + 3 + 5 + 7 + \&c. \text{ to } 2n \text{ terms} \right\} \\ & + C \frac{m^3}{2n^3} \left\{ 1^2 + 2^2 + 3^2 + \&c. \dots + 2n^2 \right\} \dagger - \frac{2n^3 m^3}{2} \\ & = 2Am + 2Bm^2 + \frac{Cm^3}{3} \left(8 + \frac{1}{n^2} \right). \end{aligned}$$

* The series $1 + 3 + 5 + 7 + 9 + \&c.$ —to $2n$ terms is thus found—

$$1 = 1^2$$

$$1 + 3 = 2^2$$

$$1 + 3 + 5 = 3^2$$

$$1 + 3 + 5 + 7 = 4^2$$

$$\&c. = \&c.$$

$$1 + 3 + 5 + \&c., \text{ to } 2n \text{ terms} = (2n)^2 = 4n^2.$$

† The sum of the series $1^2 + 2^2 + 3^2 + \&c. \dots + (2n)^2$ may be readily found by the method of Indeterminate Coefficients. Vide Wood's "Algebra."

Assume $1^2 + 2^2 + 3^2 + \&c. \dots + (2n)^2 = A + B(2n) + C(2n)^2 + D(2n)^3 + \&c.$, where A, B, C, &c., are unknown coefficients independent of $2n$; then writing $2n + 1$ for $2n$ we have $1^2 + 2^2 + 3^2 + \&c. + (2n)^2 + (2n + 1)^2 = A + B(2n + 1) + C(2n + 1)^2 + D(2n + 1)^3 + \&c.$

Subtracting the first equation from the second—

$$(2n + 1)^2 = B + C(4n + 1) + D(12n^2 + 6n + 1) + 0;$$

all the coefficients after D being separately equal to zero, since there are no terms on the left-hand side of the equation involving n^3 , or the higher powers of n .

Equating like powers of n —

$$12D = 4 \quad \text{or } D = \frac{1}{3};$$

$$4C + 6D = 4 \quad \text{or } C = \frac{1}{2};$$

$$B + C + D = 1 \quad \text{or } B = \frac{1}{2};$$

$$\text{also } A = 0.$$

$$\text{Therefore, sum required} = \frac{2m}{6} + \frac{(2m)^2}{2} + \frac{(2m)^3}{8}$$

$$\text{and sum} = \frac{(2m)^3}{2} = \frac{n^3}{3} (1 + 8n^2) = \frac{n^3}{3} \left\{ 8 + \frac{1}{n^2} \right\}$$

This last equation is true whatever n may be. When n becomes infinitely large (in which case the sum of all the trapeziums gives the exact area of the figure of which the parabolic curve $P_1 P_2 P_3$ is a boundary), we have $\frac{1}{n^2} = \frac{1}{\infty} = 0$; and the approximation to the curvilinear area $= 2 Am + 2 Bm^2 + \frac{8 Cm^3}{3}$. Putting for A , B , and C , their values in terms of a_1 , a_2 and a_3 , we have

$$\begin{aligned} \text{Area} &= 2a_1 m + (4a_2 - a_3 - 8a_1)m + \frac{8}{3}(a_3 - 2a_2 + a_1)m \\ &= \frac{m}{3} \{ 6a_1 + 12a_2 - 3a_3 - 9a_1 + 4a_3 - 8a_2 + 4a_1 \} \\ &= \frac{m}{3} \{ a_1 + 4a_2 + a_3 \}; \end{aligned}$$

which is Simpson's first rule.

SECOND DEMONSTRATION.

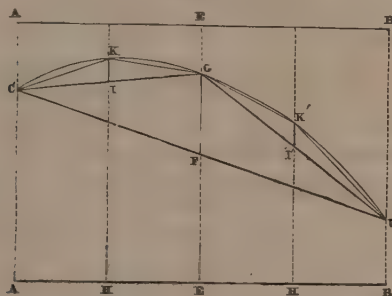
As a foundation for Simpson's First Rule of Mensuration, it might be sufficient to define a common parabola as being a curve such, that the area of any one of its segments is two-thirds of the product of the base and deflection of that segment; but it is naturally more satisfactory to most readers that this property of a parabola should be demonstrated as a consequence of some other property by the aid of which the curve can be drawn.

With that view a common parabola may be defined as follows:

A common parabola is a curve such, that the deflection of any arc is proportional to the square of its base.

In Fig. 7, CGD represents a parabolic segment, the base of which is a straight line, AB , in any convenient position perpendicular to the ordinates, AC , BD , EG , &c., on which deflections are to be

Fig. 7.



measured; the straight line, CD , is the chord of the arc, which may be either parallel or oblique to the base, AB ; and the deflection, FG , is the distance between the chord and the arc, measured on an ordinate, EG , which bisects the base, AB .

To find any number of additional points in the curve, so as to be able to draw it with any required degree of accuracy, begin by bisecting AE in H and EB in H' ; set up the ordinates, HK and $H'K'$; draw the chords, CG , GD , cutting those ordinates in I and I' , from which points lay off the deflections, IK , $I'K'$, each of which, agreeably to the definition, will be one quarter of FG ; because the bases, AE and EB , are each one half of AB ; and the square of one half is one quarter. Thus are found two new points, K and K' ; and by bisecting the bases, AH , HE , EH' , $H'B$, and proceeding as before with one quarter of the deflection, IK , or one-sixteenth of FG , four additional points may be found intermediate between the five already known: the next repetition of the process will give eight more points; and so on to any required number; the deflection of each new set of subdivisions of the curve being one quarter of that of the preceding set.

To deduce the area from this process, it is to be observed, that by taking the area of the triangle CGD , then adding those of the two triangles, CKG , $GK'D$, then those of four smaller triangles standing in the same manner on the chords CK , KG , GK' , $K'D$, and so on, a series of approximations will be got, approaching continually nearer to the exact area of the parabolic segment CGD ; and that by continuing the summation long enough, the error may be made as small as we please; a principle usually expressed by saying, that the area of the parabolic segment is the limit of the sum of the endless series of triangles above described.

But the area of the triangle CGD is one half of the product of the base and deflection of the entire segment; that is,

$$\frac{AB \times FG}{2},$$

the areas of the triangles CKG and $GK'D$ are together equal to one quarter of the area of the triangle CGD , because they stand on the same base and are of one quarter of the height; and similarly the area of each set of triangles in the series is one quarter of the area of the preceding set. Therefore the area of the parabolic segment is

$$\frac{AB \times FG}{2} \times \text{limit of the sum of the endless series,}$$

$$(1 + \frac{1}{4} + \frac{1}{16} + \frac{1}{64} + \&c.);$$

and by adding together enough of terms of that series we can get as near an approximation to the required area as we please. But the limit of the series can be found exactly as follows:

If we take the fraction $\frac{3}{4}$ of any unit, and add to it $\frac{3}{4}$ of the remaining quarter, or $\frac{3}{16}$, then $\frac{3}{4}$ of the remaining sixteenth, or $\frac{3}{64}$, and so on, we obtain successively the sums,

$$\frac{3}{4}, \frac{15}{16}, \frac{63}{64}, \&c.,$$

which obviously approach continually nearer and nearer to unity, and may be made to approach as near to unity as we please, by carrying on the series far enough. Therefore the limit of the sum of the endless series,

$$\frac{3}{4} + \frac{3}{16} + \frac{3}{64} + \&c.,$$

is unity; and dividing by 3, we find that the limit of the sum of the endless series,

$$\frac{1}{4} + \frac{1}{16} + \frac{1}{64} + \&c.,$$

is one-third. Consequently the exact area of the parabolic segment is

$$\frac{AB \times FG}{2} \times 1\frac{1}{3} = \frac{2 \times AB \times FG}{3},$$

as already stated.

Of both the preceding demonstrations it may be remarked, that they employ the reasoning of the integral calculus, though not its notation.

It being thus proved that the area of a common parabolic segment is two-thirds of the product of its base and deflection, let the figure to be measured be $ACGDB$, bounded by the base, AB , the end ordinates, AC , BD , and the convex parabolic arc, CGD . The area of this figure is made up of the trapezoid, $ACDB$, and the parabolic segment, CGD ; and the deflection, FG , of the parabolic segment is equal to the difference between the middle ordinate EG , and EF , which is the half-sum of the end ordinates. Then

$$\text{Trapezoid} = AB \times \frac{AC + BD}{2};$$

$$\text{Segment} = \frac{2}{3} AB \times \left\{ EG - \frac{AC + BD}{2} \right\}$$

Consequently, total area

$$= AB \times \left\{ \frac{2}{3} EG + \frac{AC + BD}{6} \right\}$$

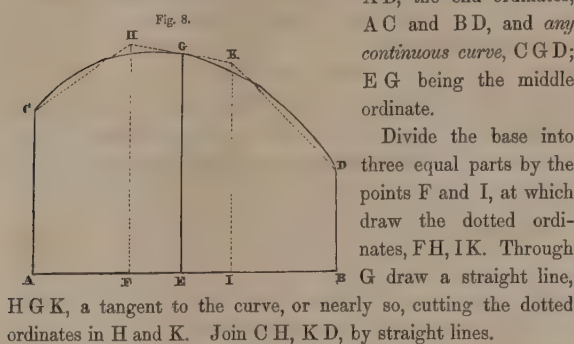
$$= \frac{AB}{6} \times \{ AC + 4EG + BD \};$$

which is Simpson's rule; because $AB + 6$ is one-third of the interval between the ordinates, AC , EG , BD .

The same result may be demonstrated when the parabolic arc is concave, by taking $A'E'B'$ parallel to AEB as the base, and finding the area $A'CGDB'$ by subtracting the parabolic segment, CGD , from the trapezoid, $A'CDB'$.

THIRD DEMONSTRATION.

Let the figure to be measured (Fig. 8) be bounded by the base,



Divide the base into three equal parts by the points F and I , at which draw the dotted ordinates, FH , IK . Through G draw a straight line, HGK , a tangent to the curve, or nearly so, cutting the dotted ordinates in H and K . Join CH , KD , by straight lines.

Then supposing it to be admitted that the series of three straight lines, CH , HK , KD , makes as good an approximation to the figure of the curve, CGD , as it is possible to make with three straight lines and no more, the area bounded by those lines, which is the sum of three trapezoids standing on the three intervals, AF , FI , IB , of the base, will be a good approximation to the area required. But by applying the rule for trapezoidal areas (Article 16) to this case, it is easily seen that the sum in question is—

$$\frac{AB}{3} \times \left\{ \frac{AC}{2} + FH + IK + \frac{BD}{2} \right\}$$

$$= \frac{AB}{6} \times \{ AC + 2FH + 2IK + BD \}.$$

Now the middle ordinate, EG , is a mean between the dotted ordinates, FH and IK , so that twice their sum is equal to four times the middle ordinate; therefore the required approximate area is—

$$\frac{AB}{6} \times \{ AC + 4EG + BD \},$$

as before.

It is not necessary for the objects of this treatise to give a detailed demonstration of Simpson's second rule; which is less used in practice than his first rule, being more complex and not more accurate. It may be remarked, however, that Mr. Barnes, in the paper already referred to, has shown how a demonstration analogous to that of Poncelet, just given, may be applied to the case of four equidistant ordinates, the multipliers for the ordinates being 1, 3, 3, 1, and the multiplier for the common interval $\frac{3}{8}$.

18A. *Subdivision of a Parabolic Figure.*—It is occasionally requisite to calculate separately the area of one of the two subdivisions (see Fig. 7) into which a parabolic figure is divided by the middle ordinate, EG . For example, supposing it is required to calculate separately the area of the subdivision $ACGE$, let AC be called the *near end ordinate*, and BD the *far end ordinate*; and the

rule will be as follows:—*To eight times the middle ordinate add five times the near end ordinate, and subtract the far end ordinate: multiply the remainder by one-twelfth of the common interval; the product will be the area required.*

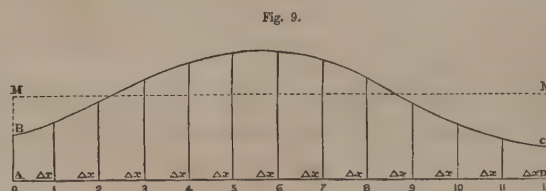
The truth of this rule is easily verified by considering that the area required is made up of the trapezoid $ACFE$, and the half-segment CGF .

[In algebraical symbols, let y be the near end ordinate, y' the middle ordinate, and y'' the far end ordinate; then,

$$\text{area} = \frac{\Delta x (5y + 8y' - y'')}{12}.]$$

19. *Areas of Arbitrary Plane Figures.*—The area of a plane figure bounded by a curved line of arbitrary form, such as a frame or a water-line of a ship, is found by measuring a sufficient number of parallel and equidistant ordinates, conceiving the figure to be divided by certain of those ordinates into figures of the parabolic kind, computing the areas of those figures, and adding them together; or else computing the sum of those areas at one operation

In Fig. 9, let $ABCD$ be the plane figure whose area is to be measured, bounded by the straight base-line or axis of abscissæ, AD ,



by two ordinates, AB and DC , at right angles to the base, and by the curved line, BC .

Divide the base into a sufficient number of equal intervals, and draw and measure ordinates at the points of division. The total number of ordinates, including the two endmost ordinates, will of course be one more than the number of intervals.

If the area is to be measured by conceiving it to be divided into trapezoids (that is, by conceiving BC to be made up of straight lines), the number of intervals into which the base is divided may be either odd or even.

If the area is to be measured by conceiving it to be divided into parabolic areas of the second order, the number of intervals must be even; if into parabolic areas of the third order, the number of intervals must be a multiple of three.

In the example represented in the figure, the base is divided into twelve equal intervals, which will suit any one of those three methods.

For the sake of uniformity in stating the rules for calculation, the ordinates which separate the parabolic areas into which the figure is conceived to be divided from each other will be called *dividing ordinates*, and all the other ordinates, except the endmost ordinates, *intermediate ordinates*.

I. **TRAPEZOIDAL RULE.**—Here every ordinate, except the endmost ordinates, is a dividing ordinate.

Add together all the dividing ordinates, and one-half of the endmost ordinates; multiply the sum by the common interval: the product will be the required area, nearly.

This is the simplest rule; but for figures whose boundaries present long sweeps of convexity or concavity, it is only a rough approximation.

II. PARABOLIC RULE OF THE SECOND ORDER, OR SIMPSON'S FIRST RULE.—Here the number of intervals must be even; and the dividing ordinates are at the distance of a double interval from each other, being those at the points 2, 4, 6, &c., in Fig. 9. The intermediate ordinates are those at 1, 3, 5, &c.

Add together the endmost ordinates, double the dividing ordinates, and four times the intermediate ordinates; multiply the sum by one-third of the common interval: the product will be the required area, nearly.

This is the most generally useful of all rules for measuring areas. It is capable of any required degree of accuracy, if the ordinates are made numerous and close enough.

III. PARABOLIC RULE OF THE THIRD ORDER, OR SIMPSON'S SECOND RULE.—Here the number of intervals must be a multiple of three; and the dividing ordinates are at the distance of a treble interval from each other, being at the points marked 3, 6, 9, in Fig. 9. The intermediate ordinates are at 1, 2, 4, 5, &c.

Add together the endmost ordinates, double the dividing ordinates, and three times the intermediate ordinates; multiply the sum by three-eighths of the common interval: the product will be the required area, nearly.

This rule is more complex than Simpson's first rule, and not more accurate.

[In algebraical symbols, those three rules for mensuration are expressed as follows:—Let L denote the whole length of the base, and n the number of intervals into which it is to be divided; then the common interval is given by the formula—

$$\Delta x = \frac{L}{n};$$

n being a multiple of 2 or 3; according to the order of the parabolic curves of which the boundary of the figure is held to consist.

Let the ordinates corresponding to the following abscissæ,

$$0, \Delta x, 2 \Delta x, 3 \Delta x, \&c. \dots n \Delta x = L,$$

be denoted as follows—

$$y_1, y_2, y_3, y_4, \&c. \dots y_{n+1}.$$

This mode of numbering the ordinates is that practised by naval architects. Amongst pure mathematicians it is more common to number them as follows—

$$y_0, y_1, y_2, y_3, \&c. \dots y_n;$$

because of the convenience of having each ordinate marked by a number proportional to the corresponding abscissa; but the former method of numbering is adopted in the following equations:—

I. TRAPEZOIDAL RULE—

$$\int y dx = \Delta x \left(\frac{y_1 + y_{n+1}}{2} + y_2 + y_3 + \&c. \right).$$

II. SIMPSON'S FIRST RULE—

$$\int y dx = \frac{\Delta x}{3} (y_1 + 4y_2 + 2y_3 + 4y_4 + 2y_5 + \&c. \dots + 2y_{n-1} + 4y_n + y_{n+1}).$$

III. SIMPSON'S SECOND RULE—

$$\int y dx = \frac{3\Delta x}{8} (y_1 + 3y_2 + 3y_3 + 2y_4 + 3y_5 + 3y_6 + 2y_7 + \&c. \dots + 2y_{n-1} + 3y_n + y_{n+1}).$$

Mathematical principles might here be explained, for determining how close together the ordinates ought to be, in order to give an approximate area of any required degree of accuracy; but it is unnecessary to do so; because the constructor, after a little experience in the use of the rules, learns to judge by the eye whether the ordinates are close enough.

Where the curved boundary of the figure is nearly at right angles to the ordinates, and where it is nearly straight, the ordinates may be far apart. Where the curved boundary is very oblique to the ordinates, and where its curvature is sharp, the ordinates must be closer together.

Much time is saved in calculation by the use of subdivided intervals, as follows:—

In those parts of the figure where close ordinates are required, the ordinary intervals may be subdivided into half-intervals, quarter-intervals, or smaller subdivisions if necessary; each ordinate belonging to a set of subdivided intervals having its multiplier diminished in the same proportion in which the intervals are subdivided. Thus, the series of multipliers for ordinates at whole intervals being—

$$1, 4, 2, 4, 2, \&c.,$$

the series of multipliers for ordinates at half-intervals will be—

$$\frac{1}{2}, 2, 1, 2, 1, \&c.,$$

and at quarter-intervals,

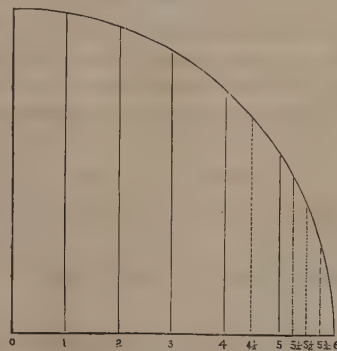
$$\frac{1}{4}, 1, \frac{1}{2}, 1, \frac{1}{2}, \&c.$$

When an ordinate stands between a larger and a smaller interval, its multiplier will be the sum of the two multipliers which it would have had as an end-ordinate for each interval. Thus, for an ordinate between a whole-interval and a half-interval, the multiplier is $1 + \frac{1}{2} = 1\frac{1}{2}$; for an ordinate between a whole-interval and a quarter-interval, $1 + \frac{1}{4} = 1\frac{1}{4}$; for an ordinate between a half-interval and a quarter-interval, $\frac{1}{2} + \frac{1}{4} = \frac{3}{4}$, &c.

The ordinates having been multiplied by their proper multipliers, and the products added together, the sum is to be multiplied by one-third of a whole-interval, to find the area.

In the following Table, those rules are applied to the calculation of the approximate area of a quadrant of a circle of 12 feet radius (Fig. 10).

Fig. 10.



This figure is chosen, because its true area to the 100th of a square foot is otherwise known to be $12 \times 12 \times .7854 = 113.10$ square feet, and this affords the means of testing the degree of approximation attained by the rules.

The ordinates are computed, to the accuracy of the 100th of a foot, by the formula, Ordinate = $\sqrt{(\text{radius}^2 - \text{abscissa}^2)}$.

In the first division of the Table, the base of the figure is divided into six intervals of two feet each. This gives an error of nearly $\frac{1}{100}$ of the whole area. In the second division of the Table the last two intervals, where the curve becomes very oblique to the ordinates, are subdivided into four half-intervals, and the error is reduced to about $\frac{1}{250}$ of the whole area. In the third division of the Table, the last two half-intervals are further subdivided into four quarter-intervals; and the error in the area becomes only $\frac{1}{800}$ of the whole.

APPROXIMATE AREA OF THE QUADRANT OF A CIRCLE OF 12 FEET RADIUS.

I. SIX INTERVALS.				II. FOUR WHOLE-INTERVALS, AND FOUR HALF-INTERVALS.				III. FOUR WHOLE-INTERVALS, TWO HALF-INTERVALS, AND FOUR QUARTER-INTERVALS.			
Number of Intervals from Origin.	Ordinates.	Multipliers.	Products.	Number of Intervals from Origin.	Ordinates.	Multipliers.	Products.	Number of Intervals from Origin.	Ordinates.	Multipliers.	Products.
0	12.00	1	12.00	0	12.00	1	12.00	0	12.00	1	12.00
1	11.88	4	47.32	1	11.83	4	47.32	1	11.83	4	47.32
2	11.31	2	22.62	2	11.31	2	22.62	2	11.31	2	22.62
3	10.39	4	41.56	3	10.39	4	41.56	3	10.39	4	41.56
4	8.94	2	17.88	4	8.94	1½	13.41	4	8.94	1½	13.41
5	6.63	4	26.52	4½	7.94	2	15.88	4½	7.94	2	15.88
				5	6.63	1	6.63	5	6.63	½	4.97
				5½				5½	5.81	1	5.81
				6	4.80	2	9.60	5½	4.80	½	2.40
								5¾	3.43	1	3.43
6	0	1	0	6	0	½	0	6	0	¼	0
× Interval,.....			167.90	× Interval,.....			169.02	× Interval,.....			169.40
3			55.96	3			56.34	3			56.46
Approximate Area,.....			111.93	Approximate Area,.....			112.68	Approximate Area,.....			112.93
True Area,.....			113.10	True Area,.....			113.10	True Area,.....			113.10
Error,.....			- 1.17	Error,.....			- 0.42	Error,.....			- 0.17
Or nearly 1/100.				Or nearly 1/250.				Or nearly 1/600.			

20. The *Mean Breadth of a Plane Figure* (or true mean of its ordinates) is the breadth of a rectangle of the same length and area with the given figure. For example, in Fig. 9, let AMND be a rectangle on the base, AD, of the same area with the curvilinear figure, ABCD; then AM is the mean breadth of that figure. The following is a rule for computing it:—

A. *Divide the area by the length: the quotient will be the mean breadth.*

[In algebraical symbols, let y_m denote the mean breadth; then

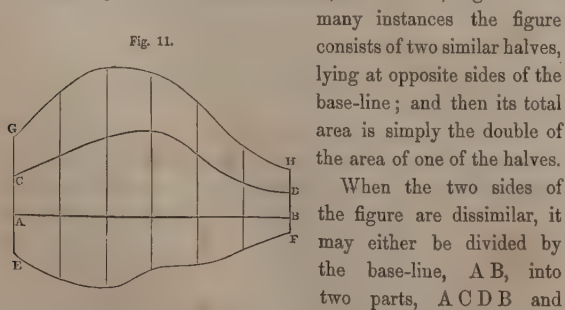
$$y_m = \frac{\int y dx}{L} = \frac{\int y dx}{n \Delta x}]$$

When the figure is computed by means of equidistant ordinates, the mean breadth may be computed irrespective of the absolute length, by the following rule:—

B. *Take the sum of ordinates and multiples of ordinates as directed in one of the rules for the computation of areas; divide by the whole number of ordinates contained in that sum (taking multiples into account): the quotient will be the mean ordinate or mean breadth of the figure.*

This process may be made use of as a step towards the computation of the area, which may be found by multiplying the mean breadth by the total length.

21. *Figures with two curved Boundaries.*—A plane figure may be bounded by curved lines at two sides, as E C D F, Fig. 11.



In many instances the figure consists of two similar halves, lying at opposite sides of the base-line; and then its total area is simply the double of the area of one of the halves. When the two sides of the figure are dissimilar, it may either be divided by the base-line, AB, into two parts, ACDB and AEFB, the areas of these parts computed separately, and then added together; or the area of the whole figure may be computed at one operation, by measuring the ordinates or breadths completely across from side to side, instead of making them terminate at the base-line; for example, EC and FD are to be taken for the endmost ordinates. In fact, the area of ECDF is equal

to the area AGHB, the dotted boundary of which, GH, is constructed by producing the ordinates of the curve, CD, and adding on to them lengths equal to the ordinates of EF, so that, for example, AG = AC + AE or CE; BH = BD + BF or DF, &c.

22. *Areas of certain special Figures.—Circle.*—The area of a circle, correct to eight places of figures, is given by multiplying the square of its radius by 3.1415926 (the common logarithm of which is 0.4971499). Convenient approximate values of that ratio are 22:7, which is too great by about 1/400 of itself, and 355:113, which is too great by about one ten-millionth of itself.

ELLIPSE.—The area of an ellipse bears the same proportion to the product of its two semi-axes which that of a circle bears to the square of its radius.

CIRCULAR SECTOR.—The area of a circular sector of a given number of degrees is found by dividing the area of the entire circle by 360, and multiplying by the number of degrees in the sector.

CIRCULAR SEGMENT.—The area of a circular segment of a given number of degrees is found by the aid of trigonometrical tables, thus—From the area of a sector of the same number of degrees, subtract the product of half the square of the radius into the sine of the given angle.

ELLIPTIC SECTOR OR SEGMENT.—The area of any sector or segment of an ellipse may be deduced from that of a corresponding

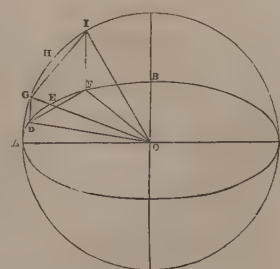
part of a circle, as follows:—Let AB (Fig. 12) be an ellipse, O its centre, OA its greater, and OB its lesser, semiaxis. About O with the radius OA describe a circle. Let it be required to find the area of the elliptic sector ODEF, or segment DEFF. Through D and F draw DG and FI, parallel to OB, cutting the circle in G and I. Join OG, OI, GI. Then,

As OA

: is to OB

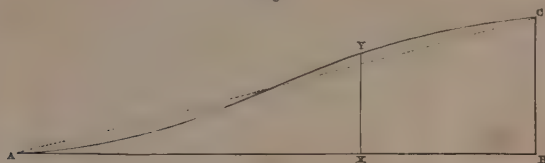
:: so is the area of the circular { sector O G H I
or segment G H I
: to the area of the elliptic { sector O D E F
or segment D E F.

Fig. 12.



CURVE OF VERSED SINES (Fig. 13).—This is a curve of the following kind:—The base, A B, and the greatest ordinate, B C, may

Fig. 13.



have any value; the ordinate, X Y, corresponding to any abscissa, A X, is given by the following proportion:—

As radius

: is to the versed sine of $\frac{AX}{AB} \times 180^\circ$

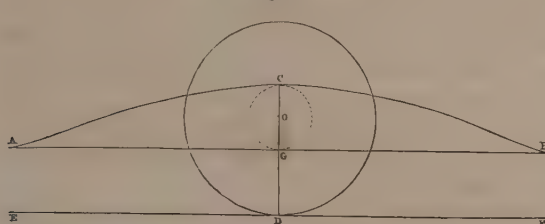
:: so is B C

: to X Y.

The area of this curve is equal to that of the triangle, A B C; that is to say, one half of the base multiplied by the extreme breadth.

TROCHOID (Fig. 14).—This curve is described by a tracing-point, C, fixed in a circular disc of the radius O D, which rolls along the straight line E F. The length of base, A B, of one complete wave

Fig. 14.



of the trochoid is equal to the circumference of the rolling circle; the extreme breadth, G C, is double of the tracing-arm, O C.

The area, A C B, is equal to the rectangle A B \times O C added to the circle of which O C is the radius.

23. Measurement of Areas by Instruments.—Instruments called "planimeters," or more correctly "platometers," are sometimes used for measuring the areas of plane figures on paper. A tracing-point is carried round the outline of the figure, and the instrument then exhibits its area by means of a graduated circle, or a dial and index. If correctly made and carefully used, those instruments are often accurate enough for many of the measurements required in designing a ship. The simplest is that of Amstler.

24. The Volumes of Solid Figures are computed as follows:—

Conceive the figure to be traversed in a convenient direction by a straight line, as base-line or axis of abscissæ, on which line divide the length of the solid into a sufficient number of equal intervals, and let the solid be conceived to be traversed by a series of plane cross sections at the points of division of the base-line. If the solid figure has flat ends perpendicular to the base-line, those ends themselves will be the endmost sections. If it is pointed, wedge-shaped, or rounded at the ends, each of the endmost sections will be 0. Measure and compute the areas of the cross sections by the rules applicable to plane figures.

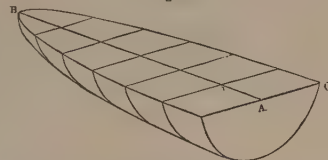
Then conceive the areas of the sections to represent the ordinates of a plane curve of the same length with the solid figure; compute the area of that ideal curve by the rules applicable to plane curves: the area so computed will be equal to the volume of the solid figure.

The curve whose ordinates represent areas of sections is sometimes drawn on paper, and is then called the "curve of areas." If drawn with sufficient accuracy, it obviously enables the volume of a figure to be found by means of the platometer.

In determining the closeness of the cross-sections, and subdividing, if necessary, the intervals between them, the same rules are to be followed as those which are applicable to the ordinates of plane figures.

EXAMPLE.—To take as an example a solid whose volume can be computed by other means—Let the figure in question be that generated by the revolution of an elliptic quadrant through a semicircle (see Fig. 15),

Fig. 15.



so that each section is a semicircle; let the length, A B, be 120 feet, divided into six intervals of 20 feet; and let the greatest half-breadth, A C, be 12 feet, being also the radius of

the greatest section. Then the area of each section is equal to $1.5708 \times$ the square of its radius; and the computation of the volume, by Simpson's first rule, is as follows:—

Number of Intervals.	Areas of Sections.	Multipliers.	Products.
0	226.20	1	226.20
1	219.91	4	879.64
2	201.06	2	402.12
3	169.65	4	678.60
4	125.66	2	251.32
5	69.12	4	276.48
6	0	1	0
			2714.36
$\times \frac{\text{Interval,}}{3}$			6 $\frac{2}{3}$
Volume,.....			18095.8 cubic ft.

This result is verified by the well-known principle that the volume of the solid now under consideration is equal to the product of its greatest sectional area into two-thirds of its length—that is to say, $226.2 \times \frac{2}{3} \times 120 = 18096$.

25. The Mean Sectional Area of a solid is found (like the mean breadth of a plane surface) either by dividing its volume by its length, or by dividing the sum of all the sectional areas and multiples of sectional areas in the calculation of the preceding article by the number of sections contained in that sum, taking account of multiples. For instance, in the example of the preceding article—

$$\frac{\text{Volume, } 18095.8}{\text{Length, } 120} = \frac{\text{Sum of sections, } 2714.36}{\text{No. of sections, } 3 \times 6 = 18} = 150.8$$

mean sectional area.

26. Direct Measurement of Volumes.—A solid, standing on a rectangular plane base, may have its volume computed directly, without the intermediate process of finding sectional areas, by the following rule, which is founded on Simpson's first rule:—

Divide each side of the rectangular base into an even number of equal intervals, and through the points of division draw a network of lines, so as to divide the base into a number of equal rectangular subdivisions. At the angles of those subdivisions measure ordinates, which multiply by the factors shown in the following table, and add the products together. Multiply the sum by one-ninth of the product of the longitudinal and transverse intervals; the product will be the volume required.

TABLE OF MULTIPLIERS FOR ORDINATES.

For 2 × 2 Subdivisions.					For 4 × 6 Subdivisions.								
1	4	1	1	4	2	4	2	4	1
4	16	4	4	16	8	16	8	16	4
1	4	1	2	8	4	8	4	8	2
					4	16	8	16	8	16	4
					1	4	2	4	2	4	1

FOR ANY NUMBER OF SUBDIVISIONS.												
1	4	2	4	2	&c.	2	4	1
4	16	8	16	8	&c.	8	16	4
2	8	4	8	4	&c.	4	8	2
4	16	8	16	8	&c.	8	16	4
2	8	4	8	4	&c.	4	8	2
&c.	&c.	&c.	&c.	&c.	&c.	&c.	&c.	&c.
"	"	"	"	"	"	"	"	"
"	"	"	"	"	"	"	"	"
"	"	"	"	"	"	"	"	"
2	8	4	8	4	&c.	4	8	2
4	16	8	16	8	&c.	8	16	4
1	4	2	4	2	&c.	2	4	1

[In algebraical symbols, let x be the abscissa, y the ordinate in the plane of the base, and z the ordinate perpendicular to the base; the volume of the solid is denoted by $\iint z \, dx \, dy$.]

28. *Measurement of Volumes in Layers, or in Rectangular Divisions.*—The volume of a solid may be computed in separate layers; the thickness of each layer being, if the trapezoidal rule is employed, one interval of the length; if Simpson's first rule, two intervals; if Simpson's second rule, three intervals. The trapezoidal rule is not always sufficiently exact; Simpson's first rule is the most generally useful. Supposing, then, that this rule is to be adopted, each layer will be bounded by two of the plane sections of the body to be measured, and will have a third plane section at the middle of its thickness. Then

I. *Add together the two outer sections and four times the middle section: one third of the sum multiplied by the interval of the sections (or half-thickness of the layer) will be the volume of the layer:—Or otherwise, one sixth of the sum will be the mean sectional area of the layer, which multiplied by its thickness, will give its volume.*

The volume of a layer whose thickness is one interval, may be computed by the rule of Article 18A.

The volume of a solid standing on a rectangular base may also be computed in separate rectangular prisms, each standing on one or more rectangular subdivisions of the base. If Simpson's first rule be taken as the foundation of the method, each prism will stand on four subdivisions of the base, measuring two longitudinal intervals lengthwise by two transverse intervals breadthwise, and will have its curved boundary defined by nine ordinates; one in the centre, one in the middle of each of the four sides, and four at the corners. Then

II. *Add together, the corner ordinates, four times the side ordinates, and sixteen times the middle ordinate; one ninth of the sum, multiplied by the longitudinal and transverse intervals, will be the volume:—Or otherwise, one thirty-sixth part of the sum will be the mean ordinate, which multiplied by the area of the base of the prism, will give its volume.*

The rectangular divisions at the edges of the base may sometimes become wedge-shaped instead of prismatic, by their outer ordinates becoming = 0.

When the volume of a solid has thus been computed in rectangular divisions, these may be added together so as to give either longitudinal or transverse layers.

29. *Volumes of certain Special Solids.*—**SPHERE.**—Multiply the cube of the diameter by .5236, or the cube of the radius by 4.1888.

ELLIPSOID.—Multiply the product of the three axes by .5236, or the product of the three semi-axes by 4.1888.

PYRAMID or CONE.—Multiply the area of the base by one-third of the height, the height being measured perpendicular to the base.

30. *Measurement of Areas by Polar Co-ordinates.*—It is occasionally requisite to measure areas, such as that shown in Fig. 16, bounded by an arc of a plane curve A B, and by two radii P A and P B, drawn to the ends of that arc from a centre or pole, P. In such cases it is convenient to use polar co-ordinates. The polar co-ordinates of a point such as B in a curve are the radius-vector P B, and the angle A P B made by that radius with some fixed direction such as P A.

In calculating areas by means of polar co-ordinates, angles are not expressed in degrees, minutes, and seconds, but in circular measure; the unit of circular measure being the angle subtended by an arc of a circle whose length is equal to the radius of the circle. The following are the factors used in converting angles expressed in degrees, minutes, and seconds, into circular measure:—

Value in Circular Measure of		Logarithms.	
360 degrees, 6.2831853	0.7981799
180 " 3.14159265	0.4971499
90 " 1.57079632	0.1961199
60 " 1.04719755	0.0200286
45 " 0.78539816	− 10 + 9.8950899
One degree, 0.01745329	− 10 + 8.2418774
One minute, 0.0002908882	− 10 + 6.4637261
One second, 0.000004848137	− 10 + 4.6855749

The following is a convenient method of approximately measuring the length of circular arcs. In Fig. 17, let O A be the radius of a circle, and A B

the arc to be measured. Draw the tangent, A D, perpendicular to O A. In A O produced, take O C = 2 O A. Join C B, and produce it till it cuts the tangent in D. A D will be nearly equal to the arc, A B. The error of this method, for an arc of 30°, is about $\frac{1}{2000}$ of the length; for larger arcs the error increases rapidly, and the method should not be used.

The following is the rule for measuring such an area as P A B in Fig. 16.

Divide the angle A P B into an even number of equal angular intervals, by means of radii P 1, P 2, P 3, &c. Measure those radii, and compute their half-squares. Treat those half-squares as if they were the ordinates of a curve, by Simpson's First Rule:—that is to say, add together the half-squares of the outermost radii, twice the half-squares of the alternate radii, and four times the half-squares of the intermediate radii; multiply the sum by one-third of the common interval, in circular measure: the product will be the area required.

Fig. 16.

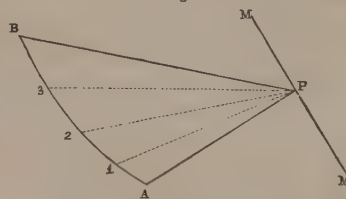
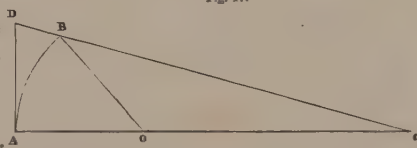


Fig. 17.



The following is an example, in which the angle APB is supposed to be $35^{\circ} 20'$, and to be divided into four equal intervals of $8^{\circ} 50'$, which, in circular measure, is equivalent to 0.15417.

Angles. With P.A.	Radii. Feet.	Half-squares of Radii.	Multipliers.	Products.
0	12.96	83.99	1	83.98
$8^{\circ} 50'$	15.12	114.31	4	457.24
$17^{\circ} 40'$	17.64	155.58	2	311.16
$26^{\circ} 30'$	20.58	211.77	4	847.08
$35^{\circ} 20'$	24.01	288.24	1	286.24
				1987.70
		$\times \frac{1}{2}$ Interval in circular measure,...		0.05139
		Area,.....		102.15 Sq.ft.

[The algebraical symbol for an area, computed by means of polar co-ordinates, is

$$\int \frac{r^2}{2} d\theta$$

where r denotes any radius of the curved boundary of the figure, and θ the angle made by that radius with one of the fixed radii which bound the figure, as P.A.]

31. *The Measurement of a Wedge-shaped Solid*, bounded by two plane surfaces meeting at a straight edge, and having cross-sections resembling Fig. 16, may be made by the following extension of the rule explained in the preceding article.

Conceive Fig. 16 to represent the wedge in question, seen end-wise; so that P represents its edge, PA and PB its two plane surfaces, and P1, P2, P3, a set of planes radiating from the edge at equal angular intervals, and making longitudinal sections of the wedge.

Treat each of the radiating plane sections, PA, P1, &c., in the following manner:—Take the straight edge, P, as above, and divide it into equal intervals. Measure ordinates as if for the purpose of taking the area of the plane figure in question; but instead of the ordinates themselves, use their *half-squares* in the calculation by Simpson's rule: the result of this calculation is called the *moment* of the given radiating plane section *about the axis*, P, for a reason which will appear in the next section.

Treat the moments thus found exactly as the half-squares of the radii are treated in Article 30; the final result of the calculation will be the volume of the wedge.

[The original investigation of this rule is in a paper by Mr. Barnes, in the "Transactions of the Institution of Naval Architects" for 1861. The algebraical expression of its result is

$$\text{Volume} = \iint \frac{r^2}{2} dx d\theta;$$

where dx denotes a small interval on the edge, P.]

SECTION II.—CENTRES AND MOMENTS OF FIGURES.

32. It will be afterwards shown, that the determination of centres of gravity, displacement, and pressure, and of moments of stability, whose importance has been explained in Chapter I., depends upon the finding of the Centres and Moments of Geometrical Figures, to which the present section relates.

33. The *Centre* of a Figure (called also the *Centre of mean distances*), whether an area or a volume, is a point such, that its distance from any given plane is the mean of the distances of all the points of that area or volume from that plane.

The centres of regular or symmetrical figures are easily found, and for the most part well known. For example, the centre of a circle, of an ellipse, of a sphere, of an ellipsoid, of a cube, &c., needs no further explanation. The centre of a parallelogram is at the

intersection of its two diagonals; the centre of a parallelopiped, at the intersection of the four diagonals which join its opposite corners.

The centres of irregular or unsymmetrical figures are found by an indirect process, which will be described further on.

34. *Geometrical Moments*.—The term "moment" is one borrowed from Mechanics. In the present section it will be considered as having a purely geometrical meaning, which is as follows:—The *moment*, or *geometrical moment*, of a figure, whether an area or a volume, *relatively to a given plane*, is the product of the magnitude of that figure into the perpendicular distance of its centre from the given plane.

When the figure under consideration is a plane area, its moment is usually taken relatively to some plane, which cuts the plane of the figure at right angles. Those two planes cut each other in a straight line, called the *axis* of moments; and it is obvious that the perpendicular distance of the centre of the figure from the plane relatively to which the moment is taken, is the same thing with the perpendicular distance of that centre from the axis of moments. The moment under consideration is, in cases of this kind, usually said to be taken *relatively to a given axis*.

35. *Principle of the Addition of Moments*.—The following principle is the foundation of all rules for finding the moments and the centres of irregular and complex figures:—

The moment of a figure relatively to a given plane (or axis) is the sum of the moments of all its parts relatively to the same plane (or axis).

Hence the process for finding the moment of an irregular figure is as follows:—Divide the figure into parts, which are either regular figures whose centres are exactly known, or figures so nearly approaching to regular figures with known centres, that their centres may be considered to be known with sufficient approximation for the purpose in view. Take the moments of all those parts, by multiplying the magnitude of each by the perpendicular distance of its centre from the given plane (or axis); the sum of those moments will be the moment of the whole figure.

When the given figure is traversed by the plane (or axis) of moments, so that some of its parts lie at one side of that plane (or axis), and some at the opposite side; then the moments of those parts which lie at one side are to be regarded as positive, and the moments of those parts which lie at the opposite side, as negative; and in computing the total moment, all the positive partial moments are to be added together, and all the negative partial moments are to be added together also; and the *difference* of those two sums is to be taken as the total moment, and is to be regarded as positive or negative according as the sum of the positive or negative moments is the greater.

In computations for practical purposes, it is advisable to avoid as much as possible the use of negative quantities; because it involves increased risk of mistakes.

It is obvious that the *moment of a figure relatively to any plane traversing its own centre is nothing*.

36. *Centre of an Irregular Figure*.—To find the distance of the centre of an irregular figure from any plane or axis, *divide the moment of that figure, relatively to the given plane or axis, by the magnitude of the figure*.

When the figure is a plane area, the centre is completely determined when its distances from two axes in the plane of the figure are known.

When the figure is a solid volume, the centre is completely determined when its distances from three planes are known.

A figure, though not regular in every respect, may be symmetrical about a certain plane, or a certain line; and then its centre lies in that plane or line.

37. *Rules for Moments and Centres of Plane Areas.*—Let the plane figure in question be such as that represented in Fig. 9, Article 19, its curved boundary being defined by means of a series of equidistant ordinates; and let it be required to find the moment of that figure relatively to a transverse axis, A B, traversing the origin; its moment relatively to the base line, A D; and the abscissa and ordinate of its centre. The rules for those purposes are as follows:

I. TO FIND THE MOMENT RELATIVELY TO THE TRANSVERSE AXIS: *Multiply each ordinate by the corresponding abscissa; treat the products as if they were the ordinates of a curve, of the same length with the given figure; the area of that curve, found by the proper rule, will be the moment required.*

II. TO FIND THE ABCISSA OF THE CENTRE: *Divide that moment by the area of the figure.*

In applying Rule I. it is often convenient to multiply each ordinate first by its proper multiplier according to Simpson's rule, and then, not by the abscissa itself, but by the number of intervals contained in the abscissa, whether integral or fractional. The sum of the products is finally to be multiplied by one-third of the square of a whole interval, to obtain the moment required.

III. TO FIND THE MOMENT RELATIVELY TO THE BASE-LINE, OR AXIS OF ABCISSÆ: *Take the half square of each ordinate; treat those half squares as if they were the ordinates of a curve, of the same length with the given figure; the area of that curve, found by the proper rule, will be the moment required.*

IV. TO FIND THE ORDINATE OF THE CENTRE: *Divide the last-mentioned moment by the area of the figure.*

[The algebraic symbols for the moments of a plane figure are as follows:—

Moment relatively to a transverse axis through the origin,

$$\int xy \, dx;$$

Moment relatively to the axis of abscissæ,

$$\int \frac{y^2}{2} \, dx.$$

Let x_0 , y_0 , denote the abscissa and ordinate of the centre; then

$$x_0 = \frac{\int xy \, dx}{\int y \, dx}; \quad y_0 = \frac{\int \frac{y^2}{2} \, dx}{\int y \, dx}.$$

EXAMPLES.

RULES I. AND II.—MOMENT OF QUADRANT OF CIRCLE RELATIVELY TO ORIGIN AT 0;

Number of Intervals from Origin.	Ordinates.	Multipliers.	Products.	Products \times Intervals from Origin.
0	12.00	1	12.00	0
1	11.83	4	47.32	47.32
2	11.31	2	22.62	45.24
3	10.39	4	41.56	124.68
4	8.94	1½	13.41	53.64
4½	7.94	2	15.88	71.46
5	6.63	½	4.97	24.85
5½	5.81	1	5.81	30.5025
6	4.80	½	2.40	13.20
6½	3.43	1	3.43	19.7225
7	0	½	0	0
				430.615
				$\times \frac{\text{Interval}^2}{8} = 1\frac{1}{2}$

(Approximate) Moment relatively to ordinate at 0,..... 574.153

Moment, 574
Area, 113 = 5.08 Approximate distance of centre of mean distances from that ordinate, or abscissa of centre.

N. B. The exact moment and distance are 576 and 5.093 respectively.

RULES III. AND IV.—MOMENT OF QUADRANT OF CIRCLE RELATIVELY TO BASE-LINE.

Number of Intervals from Origin.	Ordinates.	Half-squares.	Multipliers.	Products.
0	12.00	72	1	72
1	11.83	70	4	280
2	11.31	64	2	128
3	10.39	54	4	216
4	8.94	40	2	80
5	6.63	22	4	88
6	0	0	1	0
				864
				$\times \frac{\text{Interval}}{3} = \frac{2}{3}$
				(Exact) Moment relatively to base-line,..... 576

Moment, 576 = 5.093, ordinate from base-line to centre of mean distances.
Area, 113.1

38. *Rules for Moments and Centres of Volumes.*—

I. TO FIND THE MOMENT RELATIVELY TO THE TRANSVERSE PLANE TRAVERSING THE ORIGIN: *Multiply each sectional area by the corresponding abscissa; treat the products as if they were the ordinates of a curve of the same length with the given figure; the area of that curve, found by the proper rule, will be the moment required.*

II. TO FIND THE ABCISSA OF THE CENTRE: *Divide that moment by the volume of the figure.*

In applying Rule I. it is often convenient to multiply each sectional area, first by its proper multiplier according to Simpson's rule, and then by the number of intervals in the abscissa. The sum of the products is finally to be multiplied by one-third of the square of a whole interval.

Rules might here be added for finding the moments of a volume relatively to longitudinal planes, and the ordinates or distances of the centre from such planes, analogous to Rules III. and IV. of the preceding article; but as such rules are little if at all used in practice, it is unnecessary to give them. The usual method of finding the moment of a solid figure relatively to a plane different from that first chosen, and the distance of the centre from that plane, is to conceive the solid to be divided into layers by transverse sections parallel to the new plane of moments, and proceed as in Rules I. and II. of the present Article.

EXAMPLE.—QUARTER-SPHEROID as in Fig. 15 of Art. 24. Calculation of the moment relatively to the transverse plane traversing A, and of the distance of the centre of mean distances from that plane. Length 120 feet, in six intervals of 20 feet; radius of greatest section, 12 feet.

Number of Intervals from End Plane.	Sectional Areas.	Multipliers.	Products.	Products \times Intervals from End Plane.
0	226.20	1	226.20	0
1	219.91	4	879.64	879.64
2	201.06	2	402.12	804.24
3	169.65	4	678.60	2035.80
4	125.66	2	251.32	1005.28
5	69.12	4	276.48	1382.40
6	0	1	0	0
				6107.36
				$\times \frac{\text{Interval}^2}{3} = \frac{400}{3} = 133\frac{1}{3}$

Moment relatively to end plane at A.....814315

Moment, 814315 = 45, abscissa of centre of mean distances, being its distance Volume, 18096 from the end plane.

To find the moment relatively to the longitudinal plane through A and B, and the ordinate, or distance of the centre, from that plane, conceive the solid to be divided into six horizontal layers by horizontal plane sections at intervals of 2 feet, and proceed as

above. It is unnecessary to give the details of the calculation. The results are,

Moment, 81431.5;
Ordinate of centre, 4.5.

39. *Moments of Areas with Polar Co-ordinates.* Let the figure under consideration be a plane area like that shown in Fig. 16, Article 30, having its curved boundary defined by means of radii and angles; and let it be required to find the moment of that figure relatively to a plane, M P M, traversing the pole, P, and perpendicular to one of the radii, as P A:

I. *Divide the angle, A P B, into an even number of equal angular intervals, by means of radii, P1, P2, P3, &c. Measure those radii, and multiply the third part of the cube of each of them by the cosine of the angle which it makes with the radius, P A, which is perpendicular to the plane of moments. Treat the products of those multiplications as if they were the ordinates of a curve, by Simpson's first rule; that is to say, add together the extreme products, twice the alternate products, and four times the intermediate products; multiply the sum by one third of the common interval, in circular measure; the product will be the moment required.*

[The algebraical symbol for the moment here considered is—

$$\int \frac{r^3}{3} \cos. \theta d\theta.]$$

II. *That moment, divided by the area of the figure, will give the perpendicular distance of the centre of mean distances from the plane, M P M.*

EXAMPLE.—Total angle, A P B = 35° 20'; divided into four equal intervals of 8° 50' each, = 0.15417 in circular measure.

Radii.	Cubes of Radii. 3	Angles with P A.	Cosines.	Products.	Simpson's Multipliers.	Products.
12.96 ...	725.59 ...	0 ...	1.00000 ...	725.59 ...	1 ...	725.59
15.12 ...	1152.22 ...	8° 50' ...	0.98814 ...	1138.55 ...	4 ...	4554.20
17.64 ...	1829.68 ...	17 40 ...	0.95284 ...	1743.39 ...	2 ...	3486.78
20.58 ...	2905.46 ...	26 30 ...	0.89493 ...	2600.18 ...	4 ...	10400.72
24.01 ...	4618.76 ...	35 20 ...	0.81580 ...	3763.91 ...	1 ...	3763.91
						22931.20
Interval in circular measure, 0.15417						0.05139
x						

Moment, 1178.43

Moment, 1178.43
Area, 102.15 = 11.54, perpendicular distance of centre from the plane M P M.

III. To find the moment of the same figure relatively to one of the radii which bound it, such as P A, substitute sines for cosines in the calculation described in Rule I.

IV. Divide the moment so found by the area of the figure; the quotient will be the distance of its centre from the radius in question.

[The algebraical expression of the moment, as found by Rule III., is—

$$\int \frac{r^3}{3} \sin \theta d\theta.]$$

EXAMPLE.—The same figure as before.

Radii.	Cubes of Radii. 3	Angles with P A.	Sines.	Products.	Simpson's Multipliers.	Products.
12.96 ...	725.59 ...	0° ...	0 ...	0 ...	1 ...	0
15.12 ...	1152.22 ...	8° 50' ...	0.15356 ...	176.93 ...	4 ...	707.72
17.64 ...	1829.68 ...	17 40 ...	0.30376 ...	555.73 ...	2 ...	1111.56
20.58 ...	2905.46 ...	26 30 ...	0.44620 ...	1296.42 ...	4 ...	5185.68
24.01 ...	4618.76 ...	35 20 ...	0.57833 ...	2668.28 ...	1 ...	2668.28
						9673.24
Interval in Circular Measure,						0.05139
x						

Moment, 496.11

Moment, 496.11
Area, 102.15 = 4.86, perpendicular distance of centre from P A.

In practice it will in general be found convenient to defer the division of the cubes of the radii by 3, until after the addition of the products, and then to divide the interval by 9 instead of by 3.

40. *The moment of a wedge-formed solid,* like that described in Article 31, may be sought either relatively to a transverse sectional plane, relatively to a longitudinal plane through its edge and perpendicular to a radius, like M P M in Fig. 17, or relatively to a longitudinal plane through the edge and a radius, like P A. In the first case, the rule is the same with that given in Article 38. In the second case, the method is as follows:—

In Fig. 16, let P A, P1, P2, &c., represent longitudinal plane sections, radiating from the edge, P, as a base-line. Divide that edge into equal intervals, and treat each of the longitudinal sections as follows; measure its ordinates, compute the *third parts of their cubes*; treat those quantities as the ordinates of a curve; the area of that curve is called the *moment of inertia* of the longitudinal section in question, for a reason which will afterwards appear. Then, multiply each moment of inertia by the cosine of the angle made by the plane to which it belongs with the plane, P A, and by the multiplier corresponding to its position according to Simpson's rule; add together the products, and multiply their sum by one third of the common interval in circular measure; the result will be the moment required.

[The algebraical expression for the moment as found by this rule is—

$$\int \int \frac{r^3}{3} \cos. \theta dx d\theta;$$

$d\theta$ being an indefinitely small angular interval; dx an indefinitely small longitudinal interval on the edge, P; θ , the angle made by any given longitudinal section with P A, and r , any ordinate of a longitudinal section.]

In the third case, proceed as in the second case, multiplying by sines instead of cosines.

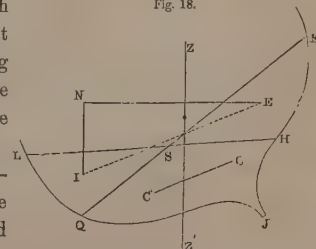
In practice it will in general be found most convenient to defer the division of the cubes of the radii by 3, until after the addition of the products, and then to divide the interval by 9 instead of by 3.

41. *Effect of shifting a Part of a Figure.*—Let K J Q, Fig. 18, represent a body of a given magnitude, whether an area or a volume; C its centre of mean distances; Z Z a plane, with respect to which the moment of the body is known, being the product of its magnitude into the perpendicular distance of C from that plane.

Suppose that without altering the total magnitude of the figure, a portion of it is shifted into a new position: for example, suppose the portion, H S K, to be taken away from its original position, and set on in the new position, Q S L, so as to alter the shape of the body from Q J K to L J H; and let it be required to find the effect of that change upon the moment of the body relatively to the plane Z Z, and upon the position of its centre of mean distances. Let E be the original and I the new position of the centre of the shifted portion of the body:—

I. *The alteration of the moment of the body relatively to the plane Z Z, is found by multiplying the magnitude of the part shifted by the*

Fig. 18.



distance through which the centre of that part is shifted in a direction perpendicular to the plane ZZ.

That distance is found as follows. Through E draw EN perpendicular to the plane ZZ, and through I draw IN parallel to the same plane, cutting EN in N; EN will be the distance in question.

II. To find the new position of the centre of the entire body: through the original centre, C, draw CC' parallel to EI, and bearing to that line the following proportion—

As the magnitude of the entire body,
: is to the magnitude of the part shifted,
: so is EI.
: to CC';

then C' will be the new centre of mean distances of the entire body.

42. Centres and Moments of certain special Figures.

I. TRIANGLE.—Fig. 19; the centre, G, is at the point of intersection of the three straight lines, AD, BE, CF, drawn from the angles to the centres of the opposite sides; and it cuts off one-third part from each of those lines.

II. TRAPEZOID.—Fig. 20, or quadrilateral with two sides parallel. Let AB, CD, be the parallel sides. Bisect them in E and F; join EF. Then—

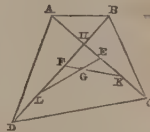
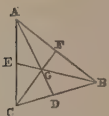
$$EG = \frac{EF}{2} \left\{ 1 - \frac{1}{3} \cdot \frac{CD - AB}{CD + AB} \right\}$$

and G will be the centre.

Fig. 19.

Fig. 20.

Fig. 21.

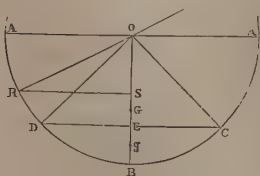


III. TRAPEZIUM, or irregular quadrilateral (Fig. 21), ABCD. Draw the diagonals, AC, BD, cutting each other in H. Bisect them in E, F. Make DL = HB; CK = AH. Join KF, LE; their intersection, G, will be the centre required.

IV. PLANE POLYGON.—Any plane figure bounded by straight lines may be divided into triangles; when the moments of those triangles relatively to a given plane, being added together, will give the moment of the entire polygon relatively to that plane; and that moment being divided by the area of the polygon will give the distance of its centre from the same plane.

V. CIRCULAR SECTOR ODBC (Fig. 22).—Draw OB, bisecting the angle DOC; the centre of mean distances will be in that line.

Fig. 22.



To find its distance, OG, from the centre of the circle: multiply two-thirds of the radius by the chord CD, and divide by the length of the arc CBD.

If the chord is taken from trigonometrical tables (being twice the sine of half the

angle, DOC), the arc must be expressed in circular measure. (See Art. 30.)

The moment of the sector relatively to a plane, AOA, perpendicular to the centre line, OB, and traversing the centre of the circle, is one-third of the square of the radius multiplied by the chord CD (the actual length of the chord being here understood).

The moment of the same sector relatively to a plane, OR, traversing the centre of the circle in a direction oblique to OB, is found by multiplying the before-mentioned moment by the sine of the angle BOR; that is, by the ratio $\frac{RS}{OB}$; RS being perpendicular to OB.

[In algebraical symbols, let the radius OB = r; $\frac{DOC}{2} = DOB = \phi$; BOR = θ ; then

$$OG = \frac{2r \sin \phi}{3 \phi};$$

$$\text{Moment relatively to AOA} = \frac{2r^3 \sin \phi}{3};$$

$$\text{Moment relatively to OR} = \frac{2r^3 \sin \phi \sin \theta}{3}.]$$

VI. CIRCULAR SEGMENT, CBD (Fig. 22). The moment relatively to AOA is one-twelfth of the cube of the chord CD; or in other words, two-thirds of the cube of the half chord CE.

The moment relatively to an oblique plane, OR, is found by multiplying the before-mentioned moment by the sine of the angle BOR; that is, by $\frac{RS}{OB}$.

The centre of mean distances is in the line OB which bisects the arc CD. To find its distance Og, from the centre of the circle, divide the moment of the segment relatively to AOA, by its area (for which see Art. 22).

[Using the same symbols as in the case of the sector, we have,

$$\text{Moment relatively to AOA} = \frac{CD^3}{12} = \frac{2r^3 \sin^3 \phi}{3};$$

$$\text{Moment relatively to OR} = \frac{CD^3 \cdot RS}{12 OB} = \frac{2r^3 \sin^3 \phi \sin \theta}{3};$$

$$Og = \frac{2r \sin^3 \phi}{3(\phi - \sin \phi \cos \phi)} = \frac{2r \sin^3 \phi}{3(\phi - \frac{\sin 2\phi}{2})};$$

ϕ being expressed in circular measure.]

VII. ELLIPTIC SECTORS AND SEGMENTS.—(Fig. 23.) Let ABA be an ellipse; O its centre; AOA its greater axis, Ob its lesser semi-axis; and let it be required to find the centre of mean distances of the sector, Oced, or of the segment, cde.

On the diameter, AOA, describe the circle, ABA; draw dD and cC parallel to OB, cutting that circle in D and C; then

OCD will be a circular sector, and CDE a circular segment, corresponding to the elliptic sector and segment respectively (as in Art. 22). Bisect the circular sector by the radius, OE; draw Ee parallel to OB, cutting the ellipse in e; join Oe; this will be the central radius of the elliptic sector and segment, in which their centres of mean distances will be situated; and those centres will divide that radius in the same proportions in which the centres of mean distances of the corresponding circular sector and segment divide their central radius, OE.

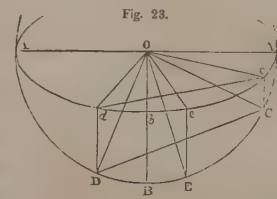
The moments relatively to AOA and OB are given by the following proportions:—

MOMENTS RELATIVELY TO AOA.

$$OB^3 : Ob^3 :: \text{Moment of circular } \left\{ \begin{array}{l} \text{sector.} \\ \text{segment.} \end{array} \right\} : \text{Moment of elliptic } \left\{ \begin{array}{l} \text{sector.} \\ \text{segment.} \end{array} \right\}.$$

MOMENTS RELATIVELY TO OB.

$$OB : Ob :: \text{Moment of circular } \left\{ \begin{array}{l} \text{sector.} \\ \text{segment.} \end{array} \right\} : \text{Moment of elliptic } \left\{ \begin{array}{l} \text{sector.} \\ \text{segment.} \end{array} \right\}.$$



VIII. CONE OR PYRAMID.—Find the centre of the base, from which draw a straight line to the summit; this will be the axis of the cone or pyramid. The centre of mean distances is in the axis, at one-fourth of its length from the base.

IX. HEMISPHERE, OR HEMI-ELLIPSOID.—The distance of the centre of mean distances from the circular or elliptic base is three-eighths of the radius of the sphere, or of that semi-axis of the ellipsoid which is perpendicular to the base.

43. *Geometrical Moments of Inertia*.—The term "moment of inertia," like the term "moment," is borrowed from Mechanics, and its mechanical use will be explained in a later section. In the geometrical sense, its meaning is as follows:—

'Suppose any space of a given figure and magnitude to be divided into an indefinitely great number of indefinitely small parts; that the magnitude of each of those small parts is multiplied by the square of its perpendicular distance from a given axis, and that all the products so obtained are added together; the sum is the *moment of inertia of the given space about the given axis*.

If the moment of inertia of a given figure about a given axis be divided by the magnitude of that figure, the quotient is what is called the *square of the radius of gyration* of that figure about the given axis; the radius of gyration of a figure about a given axis being a line, the square of whose length is the *mean of the squares of the distances of all parts of the figure from the given axis*.

[The algebraical symbol for the moment of inertia of a plane area is—

$$\iint r^2 dx dy,$$

where dx and dy represent the two dimensions of one of the small parts into which the area is divided, and r the perpendicular distance of that part from the given axis.

In the case of a volume, the corresponding symbol is—

$$\iiint r^2 dx dy dz.$$

The square of the radius of gyration of a plane area is expressed by—

$$\frac{\iint r^2 dx dy}{\int y dx},$$

and that of a volume by—

$$\frac{\iiint r^2 dx dy dz}{\int z dx dy}.$$

The moments of inertia which are of most frequent use in naval architecture are those of arbitrary plane figures about their base lines, and are found by the aid of the following principle:—

I. *The moment of inertia of a rectangle about its base is equal to the length of the base multiplied by one-third of the cube of the height of the rectangle.* In other words, the square of the radius of gyration of a rectangle about its base is one-third of the square of its height.

[In algebraical symbols, let x denote the base, and y the height; then—

$$\text{Moment of inertia} = \frac{xy^3}{3}; \text{ and}$$

$$(\text{radius of gyration})^2 = \frac{y^2}{3}.]$$

The following is the rule for finding the moment of inertia of a plane area, such as that represented in Fig. 9, Art. 19, about its base line:—

II. *Divide the base into a suitable number of equal intervals, and measure ordinates at the points of division; take the third part of*

the cube of each of those ordinates, and treat the quantities so computed as the ordinates of a new curve; the area of that new curve, found by one of the rules for that purpose, will represent the moment of inertia of the original figure about its base line.

[In algebraical symbols, the moment of inertia of a plane area about the axis of abscissæ is represented by—

$$\int \frac{y^3}{3} dx.]$$

When the moment of inertia is required as a whole, and not in separate parts, time is saved by postponing the division of the cubes by 3 until the end of the calculation, when it may be done once for all.

Should it be desired to find the moment of inertia of a plane figure relatively to one of its ordinates, the following is the rule to be observed:—

III. *Multiply each ordinate by its proper multiplier according to Simpson's rule; then multiply each of the products by the square of the number of whole intervals that the ordinate in question is distant from the ordinate taken as an axis of moments: add together the products, and multiply their sum by one-third of the cube of a whole interval; the product will be the moment of inertia required.*

[In algebraical symbols this is expressed by—

$$\int x^2 y dx.]$$

EXAMPLES.—As in Art. 19, Fig. 10, let the figure chosen be a quadrant of a circle of 12 feet radius, the base divided into six intervals of 2 feet. In the second of the following calculations, some of the intervals are subdivided into halves and quarters.

RULE II.—CALCULATION OF MOMENT OF INERTIA RELATIVELY TO THE BASE.

Number of Intervals.	Ordinates.	Cubes ÷ 3.	Multipliers.	Products.
0	12.00	576.00	1	576.00
1	11.83	552.16	4	2208.64
2	11.31	482.73	2	965.46
3	10.39	374.11	4	1496.44
4	8.94	238.51	2	477.02
5	6.63	97.28	4	389.12
6	0	0	1	0
				6112.68
				$\times \frac{\text{Interval}^3}{3}$
				4075.12
				Approximate moment of inertia,.....
				4075.12
				36.03 approximate square of radius of gyration.
				Area, 113.1 = 36.00 true square of radius of gyration, otherwise known.
				Error, + 0.03

RULE III.—CALCULATION OF MOMENT OF INERTIA RELATIVELY TO ENDMOST ORDINATE.

Number of Intervals.	Ordinates.	Simpson's Multipliers.	Products.	Squares of numbers of Intervals.	Products
0	12.00	1	12.00	0	0
1	11.83	4	47.32	1	47.32
2	11.31	2	22.62	4	90.48
3	10.39	4	41.56	9	374.04
4	8.94	1½	13.41	16	214.56
4½	7.94	2	15.88	20¼	321.57
5	6.63	¾	4.97	25	124.25
5½	5.81	1	5.81	27⅞	160.14
5¾	4.80	¾	2.40	30¼	72.60
5¾	3.43	1	3.43	38⅞	113.40
6	0	¼	0	36	0
					1518.36
				$\times \frac{\text{Interval}^3}{3}$	22½
					4048.96
					Approximate moment of inertia,.....
					4048.96
					35.8 approximate square of radius of gyration.
					Area, 113.1 = 36.0 true square of radius of gyration, otherwise known.
					Error, — 0.2

The true moment of inertia, about either of the axes chosen, is 4070 (neglecting fractions).

IV.—To find the difference between the moments of inertia of the same figure about two axes parallel to each other: multiply the magnitude of the figure (area or volume as the case may be) by the difference between the squares of the distances of the two axes from the centre of mean distances of the figure. The moment of inertia will be the greater about that axis which is farther from the centre. In other words, the difference of the squares of the radii of gyration about the two axes is equal to the difference of the squares of their distances from the centre.

EXAMPLE (1). The square of the radius of gyration of a triangle about its base, is one-sixth of the square of its height. What is the square of the radius of gyration of the same triangle about an axis parallel to its base, traversing its centre?

Square of original radius of gyration,	$\frac{1}{6} (\text{height})^2$
Subtract square of its distance from centre,	$\frac{1}{3} (\text{height})^2$
Square of new radius of gyration,	$\frac{1}{6} (\text{height})^2$

(2.) Required, the square of the radius of gyration of the same triangle about an axis parallel to its base, traversing its summit.

Square of original radius of gyration,	$\frac{1}{6} (\text{height})^2$
Square of distance of base from centre,	$\frac{1}{3} (\text{height})^2$
Square of distance of summit from centre,	$\frac{1}{3} (\text{height})^2$
Difference, to be added,	$\frac{1}{3} (\text{height})^2$
Square of new radius of gyration,	$\frac{1}{3} (\text{height})^2$

Each of those squares of radii of gyration, being multiplied by the area of the triangle, gives the corresponding moment of inertia.

The use of the moments of inertia of plane figures in computing the moment of a wedge relatively to a plane, has been stated in Article 40. Other uses will be explained in the sequel of this treatise.

44. *Moments of Inertia and Radii of gyration of Special Figures.*—The moments of inertia of similar plane figures are proportional to their lengths (parallel to the axis) and to the cubes of their breadths, and may be computed in each case by multiplying the product of the length and cube of the breadth by a factor (always fractional) depending on the figure. The squares of the radii of gyration of similar figures are proportional to the squares of their breadths (perpendicular to the axis), and may be computed in each case by multiplying the square of the breadth by a factor (always fractional) depending on the figure.

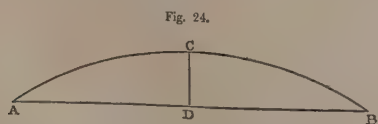


Fig. 24.

The following are the cases to which reference will have to be made in the sequel:—

Figure.	Axis.	Moment of Inertia = length \times breadth ³ \times	(Radius of Gyration) ² = breadth ² \times
I. Rectangle,	Base, ...	$\frac{1}{3}$	$\frac{1}{3}$
II. Triangle,	— ...	$\frac{1}{8}$	$\frac{1}{8}$
III. Circle, or Ellipse,	Any diameter, ...	$\frac{7854}{16} = .491$ nearly	$\frac{1}{8}$
IV. “ “	Tangent, ...	$\frac{5 \times 7854}{16} = .2454$ nearly	$\frac{1}{8}$
V. Curve of Versed sines, as in Fig. 18, Art. 22, }	Base A B ...	$\frac{2}{3}$	$\frac{2}{3}$
VI. Common Parabola, as in Fig. 24, vertex C at extreme breadth, ... }	Base A B ...	$\frac{1}{105}$	$\frac{3}{35}$
VII. Trochoid, as in Fig. 14, Art. 22,	Base A B ...	$\left\{ \frac{2}{3} (\text{length} \times \text{breadth}^3) \right\}$ $\left\{ + .2454 \text{ breadth}^4 \right\}$	—

SECTION III.—SUMMARY OF MECHANICAL PRINCIPLES.

45. *Units of Force.*—The standard unit for the measurement of forces in Britain is the *pound avoirdupois*. It is convenient, however, in calculations respecting the hydraulics of shipbuilding, to use as occasional units of force the *ton* of 2240 lbs., and the weight of a *cubic foot of sea-water*, being the thirty-fifth part of a ton, or 64 lbs. avoirdupois. The last of those units is the most convenient in calculations respecting the buoyancy and stability of ships; the pound avoirdupois is the most convenient in calculations respecting the engine-power required to drive them.

The standard unit of force in the metrical system is the kilogramme; it is sensibly equal to the weight of one-thousandth part of a cubic metre of pure water. The metrical ton weight, or *tonneau*, is 1000 kilogrammes, or (sensibly) the weight of a cubic metre of pure water.

COMPARISON OF BRITISH AND METRICAL UNITS OF WEIGHT.

	Ratios.	Logarithms.	Logarithms.	Ratios.
Pounds avoirdupois } in a kilogramme, }	2.20462	0.3433340	1.6566660	0.453593 { Kilog. in a lb. avoird.
Ton in a tonneau,	0.984206	1.9930860	0.0069140	1.01605 tonneaux in a ton.

46. *Units of Statical Moment, or of Couples.*—As already stated in Chap. I., Art. 4, if a pair of equal and opposite forces are applied to one body, in directions which, though parallel, are not directly opposed, their tendency is to turn the body about. For example, in Fig. 25, the arrows, AB and CD, represent a pair of equal forces, which, if they were directly opposed to each other, would exactly balance each other; but as their directions, though parallel and opposite, are not directly opposed, their tendency is to turn the body on which they act in the direction represented by the curved arrow. Such a pair of forces is called a *couple*, or *statical couple*. Such a couple as that represented in the figure, which tends to produce rotation in the direction opposite to that of the hands of a watch, is called *left-handed*; a couple tending to produce rotation in the direction of the hands of a watch, is said to be *right-handed*. A man pulling with his right hand, and pushing with his left against the two ends of the cross-head of an auger, exerts a right-handed couple; when he pulls with his left hand, and pushes with his right, he exerts a left-handed couple.

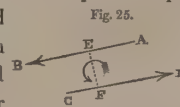


Fig. 25.

The tendency of a couple to turn the body on which it acts is expressed as a quantity by multiplying the common magnitude of the two equal forces which make the couple, by the perpendicular distance between the lines in which they act, which perpendicular distance is called the *arm* of the couple. For example, the arm of the couple in Fig. 25 is EF, the perpendicular distance between the lines of action of the two forces. In the case used as an example, of a man driving an auger, the arm of the couple is the distance between his hands.

The product of the force by the arm is called the *moment* of a couple. The standard unit of moment of a couple, in British measures, is the moment of a couple of forces, each of one avoirdupois pound, with an arm of one foot; and it is called a *foot-pound*.

Other units are occasionally used, even in British measures. For example, the arm of a couple may be expressed in inches, and the force in pounds; and in performing the multiplication, the moment will be expressed in *inch-pounds*, of which twelve make a foot-pound. Or the arm may be expressed in feet, and the force in tons; and then the moment will be expressed in *foot-tons*, one of which is equal to 2240 foot-pounds, or to 26,880 inch-pounds.

In calculations respecting the stability of ships, the moments of couples may occasionally be expressed by multiplying the force, in cubic feet of sea-water, by the arm in feet; in such cases the moment is expressed in units, each of which is equal to 64 foot-pounds, and of which 35 make a foot-ton.

The ordinary unit of statical moment in the metrical system of measures is the *kilogrammetre*, being the moment of a couple of forces each of one kilogramme, with an arm of one metre. The proportions between its value and that of a foot-pound are as follows:—

	Ratio.	Logarithm.	Logarithm.	Ratio.
Foot-pounds in a kilogrammetre,	7.23314	0.8593269	1.1406731	0.138253
	{ Kilogrammetre in a ft.-pound.			

47. *Units of Mechanical Work and Energy.*—A British unit of mechanical work consists in the overcoming of a resistance of one pound avoirdupois through the distance of one foot; the distance being measured in the direction exactly opposite to that in which the resistance acts. For example, in lifting a weight of one pound to a height of one foot, one British unit of work is done. *Energy* means the power to perform work, and is measured in the same sort of units with work. The unit of work just described is called a *foot-pound*, being the same name which is applied to an unit of statical moment; but the quantities denoted are altogether different.

The quantity of work done in overcoming a given resistance through a given distance, is computed in foot-pounds, by multiplying the resistance in pounds by the distance in feet.

In the metrical system, an unit of work consists in overcoming one kilogramme of resistance through the distance of one metre. It is called a *kilogrammetre*, being the same name that is applied to the unit of statical moment; but the quantities denoted are of wholly distinct kinds.

48. *Units of Speed.*—In British scientific calculations connected with mechanics, the standard unit of speed is the *foot per second*. The *foot per minute* is also sometimes used, and of these of course 60 make one foot per second.

The speed of ships is sometimes stated in statute miles per hour, and sometimes in nautical miles per hour. The statute mile is 5280 feet; the nautical mile, or *knot*, is the mean length, on the earth's surface, of one minute of a meridian. The earth's dimensions and figure are not yet known with sufficient precision to enable the length of the nautical mile to be stated with any great exactness according to this definition; it may be taken, however, for practical purposes, at 6076½ feet.

The following table shows a comparison between different measures of speed.

	Statute Miles per Hour.	Feet per Second.	Feet per Minute.	Feet per Hour.
1	1.46	88	5280
0.6818	1	60	3600
0.01136	0.016	1	60
0.0001893	0.00027	0.016	1
One nautical mile, or knot, per hour,	1.1509	1.6879	101.275	6076½

49. *Units of Speed of Turning.*—The speed with which a body turns may be expressed in turns per second, or per minute, or per hour. But for certain mechanical purposes, the best mode of expressing *angular velocity*, as the speed of turning is called, is in *units of circular measure per second* (see Art. 30). To reduce *turns per second* to *units of circular measure per second*, multiply by the ratio of the circumference of a circle to its radius, viz.—6.2831853. The meaning of an angular velocity expressed in units of circular measure per second may be explained by saying, that it is the

distance in feet run through in one second by a point whose distance from the axis of rotation is one foot.

50. *Intensity of a Distributed Force.*—As stated in Chap. I., Art. 3, every force has its action spread over a certain space—an area, or a volume, as the case may be.

By the *intensity* of a force is to be understood a quantity found by dividing the amount of the force by the extent of the space over which its action is distributed: so that conversely, the intensity, multiplied by the extent of the space, gives the amount of the force.

The only force distributed through a volume whose action will have to be considered for the present, is that of gravity. The weight of any body at the earth's surface is a force acting between the entire mass of the body and the entire mass of the earth, and its action on the body is spread over the whole volume of the body. If we divide the weight of a body, in pounds avoirdupois, by its bulk in cubic feet, we obtain a quantity which might be called the *intensity of the weight of that body in pounds to the cubic foot*, and which is, in fact, the weight of one cubic foot of the material of which the body consists. Notwithstanding the frequent practical use of quantities of this sort, there is no single word at present in general use to denote them. In some lately published writings, the word *heaviness* has been proposed for that purpose, as being a convenient term, and one which has never before had a definite scientific meaning assigned to it.*

The *unit of heaviness*, in ordinary British standard measures, is *one pound avoirdupois to the cubic foot*; the following are some examples of its use:—

	Lbs. to the Cubic foot.
Heaviness of Wrought-iron,	480
" British Oak Timber,	55
" Norway Red Pine Timber,	36
" Pure Water,	62.4
" Sea-water,	64

Many other examples will be given in the tables contained in Division Third of this treatise.

Another sort of unit of the intensity of weight is the weight of a given bulk of some standard substance; and for this purpose pure water has been chosen. The term *specific gravity* denotes the proportion which the weight of a given body bears to the weight of the same bulk of pure water. For exact scientific purposes, it is necessary to fix exactly the temperature of the water with which the body is compared; in Britain, the temperature so fixed is 62° Fahr.; in countries where the metrical system prevails, it is the temperature at which a given weight of water contracts to its least volume—viz., 39.1° Fahr., or 3.94° Cent. The weight of a cubic foot of water is—

	Lbs. Avoirdupois.
At 39.1° Fahr.	62.425
At 62° "	62.355

For ordinary practical purposes, it is unnecessary to attend to that difference; and the weight of one cubic foot of pure water may be taken at 62.4 lbs. Hence the following rules:—

I. To find the *HEAVINESS* of a substance, in pounds avoirdupois to the cubic foot, multiply its *specific gravity* by 62.4.

II. To find the *SPECIFIC GRAVITY* of a substance, divide its *heaviness*, in pounds avoirdupois to the cubic foot, by 62.4.

For ease of calculation, where precision is not required, 62.5, being = $\frac{1000}{16}$, may be used instead of 62.4.

* Rankine on Civil Engineering, p. 151.

COMPARISON OF BRITISH AND METRICAL UNITS OF HEAVINESS.

	Ratio.	Logarithm.	Logarithm.	Ratio.
Pounds to the cubic foot in a kilo. to the cubic metre.	0.062425	2.7953558	1.2046447	16.019
Kilos. to the cubic metre in a pound to the cubic foot.				

51. *Intensity of Pressure.*—Pressure or thrust, and tension or pull, are forces distributed over surfaces. The special consideration of tension and of oblique pressure may be deferred until that division of this treatise which relates to the strength of materials; for the present our attention will be confined to pressures like those exerted between a fluid and the surface of a solid body, or by the parts of a fluid against each other, which pressures are exerted at right angles to the surfaces where they act.

The *intensity of a pressure* is found by dividing its amount by the *area* of the surface over which it is distributed; and conversely, the intensity multiplied by the area gives the amount. For example, if the amount of a pressure be expressed in pounds, and the area in square feet, the division of the former by the latter gives the intensity in pounds on the square foot; and the multiplication of the area in square feet by the intensity in pounds on the square foot, gives the amount of the pressure in pounds.

Thus the standard British unit of intensity of pressure is the *pound avoirdupois on the square foot*. Other units are often used as being more convenient for special purposes; for example, the *pound on the square inch, equivalent to 144 pounds on the square foot*.

The most convenient mode, however, of expressing the intensity of a pressure for purposes connected with hydraulics, is *feet of depth of water*. The surface at which the pressure acts being conceived to be horizontal, let a column of water be supposed to stand upon it, of a weight equal to the amount of the pressure; then the vertical depth of that column will represent the intensity of the pressure, at the rate of one foot of depth of pure water for every 62.4 lbs. of intensity to the square foot, or one foot of depth of sea-water for every 64 lbs. of intensity to the square foot.

TABLE OF COMPARISON OF UNITS OF INTENSITY OF PRESSURE.

Feet of Sea-water.	Feet of Pure Water.	Lbs. on the Square Foot.	Lbs. on the Square Inch.	Inches of Mercury at 32°.
1.105	1.1884	70.73	0.4912	1
1	1.026	64	0.444	0.905
0.975	1	62.4	0.438	0.8823
0.015625	0.016026	1	0.00694	0.01414
2.25	2.308	144	1	2.036

TABLE OF COMPARISON OF BRITISH AND METRICAL UNITS OF INTENSITY OF PRESSURE.

	Ratios.	Logarithms.	Logarithms.	Ratios.
Pounds on the square foot in a kilo. on the square metre.	0.2048098	1.3113482	0.6886518	4.88261
Pounds on the square inch in a kilo. on the square millimetre.	1422.28	3.1529858	4.8470142	0.000708095
Kilos. on the square metre in a pound on the square inch.				

52. The *Equality of Action and Reaction* denotes a physical law, or universal fact, which may be stated as follows—*Every force consists in a pair of equal and opposite actions between a pair of bodies.*

The form which this law takes in regard to terrestrial gravity—viz., that a body on the earth's surface exerts an attraction on the earth equal and opposite to the attraction which the earth exerts on the body—has no special application to questions of practical mechanics.

In regard to pressures, on the other hand, its applications are very important; for example, the vertical pressure exerted between the water and the bottom of a ship consists in a downward pressure of the ship on the water, and an exactly equal and opposite upward pressure of the water on the ship: the horizontal pressure exerted between the propeller of a steam-vessel and the water consists in a backward pressure of the propeller on the water, and an exactly

equal and opposite forward pressure of the water on the propeller, which forward pressure is transmitted through the propeller and the bearings of its shaft to the ship: the pressure exerted between the wind and a sail consists in a pressure of the sail against the air, and an exactly equal and opposite pressure of the air against the sail. These facts have already been occasionally referred to in the course of Chap. I.

53. *Balance of Forces Acting through one Point.*—The effect of a force upon one of the two bodies between which it acts may be of one or other of two kinds—to balance another force or forces, or to produce or modify motion in the body. In the present article, we shall attend only to effects of the first kind.

The principles of the balance or equilibrium of forces which act through one point depend upon one principle—that *any force acting through a point may be represented by a line*; and that *the result of combining forces may be represented by putting together the lines which represent the forces.*

A force is represented by a line in the following manner:—Let A be a point through which a force acts, of a given amount, and in a given direction. Through the point, A, a line, BC, is drawn in the given direction, and of a length representing the magnitude of the force, according to any convenient scale.

An arrow-head (at C) points out towards which end of the line the force tends to produce motion. Then the line, BC, completely represents the given force.

Fig. 26.



It is in most cases convenient, though not absolutely essential, to make the line start from the point, A; that is, to make B coincide with A.

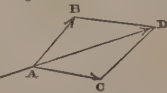
How great soever the number of forces which act through one point, it is always possible to find, by putting together the lines which represent them, a single force which shall be equivalent to them all. That single force is called the *RESULTANT* of all the combined forces, and is equal and opposite to the force required in order to balance them. Thus all questions as to the equilibrium of the forces acting through one point depend upon finding the resultant of a system of forces; for if that resultant proves to be nothing, the forces are already balanced; and if it proves to be something, a force exactly equal and opposite to that resultant is what is wanted in order to produce equilibrium.

The following are problems as to the resultants of forces acting through one point, to which there will be occasion to refer in the sequel.

I. *The Resultant of a set of Forces acting along one line, is their Algebraical Sum.*—By taking the algebraical sum of the forces is meant, that those which act in one direction (it matters not which) are to be considered as positive, and those which act in the contrary direction as negative. The arithmetical sum of all the positive forces is to be taken; also the arithmetical sum of all the negative forces. The direction of the resultant (positive or negative) will be that of the greater of those two sums; and its magnitude, the difference between them.

II. *PARALLELOGRAM OF FORCES.*—(Fig. 27.)—To find the resultant of two forces which act through one point, A, along different lines, let AB and AC represent those forces; through B draw BD parallel to AC; through C draw CD parallel to AB; the diagonal, AD, of the parallelogram will represent the resultant.

Fig. 27.

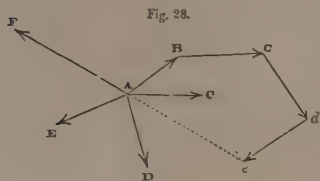


A E, equal and directly opposite to A D, represents the force required in order to balance A B and A C.

It is evident that three forces, A B, A C, and A E, which balance each other at one point, act in one plane; and that their magnitudes are proportional to the sides of a triangle, A B D, which are respectively parallel to their directions. That is to say, *each force is proportional to the sine of the angle between the directions of the other two*. Thus all questions as to the balance of forces acting through one point may be reduced to questions of trigonometry.

The "parallelogram of forces" is a particular case of a more comprehensive proposition, viz.:

III. POLYGON OF FORCES.—(Fig. 28).—To find the resultant of



any number of forces acting through one point along different lines, represent the forces in magnitude and direction by lines starting from the given point; then taking any one of those lines to begin with, put together, end to end, a series of lines equal and parallel to those representing the forces; draw a straight line from the starting-point to the end of that series of lines; that straight line will represent the required resultant.

A line equal and directly opposite to that line will, of course, represent the force wanted in order to make equilibrium; and if the series of lines comes round again to the starting-point, so as to make a closed polygon, the resultant is nothing, and the forces are already balanced.

For example, let the given forces be four in number, and represented by A B, A C, A D, A E. Choosing any one of them to begin with, for example A B, from B draw B c parallel and equal to A C; from c draw c d parallel and equal to A D; from d draw d e parallel and equal to A E. This is what is meant by putting together a series of lines parallel and equal to these representing the forces. Then from the starting-point, A, to the termination, e, of the series of lines, draw the straight line, A e; this will represent the resultant; and a line, A F, equal and directly opposite to A e, will represent the force wanted in order to complete the equilibrium. Suppose now that the force, A F, is actually combined with the others—then upon joining at e, to the series of lines already drawn, a line parallel and equal to A F, that line (viz., e A) is found to lead exactly back to the starting-point, A, so as to form a closed polygon; showing that the resultant is now nothing, and the equilibrium complete.

The forces, in this problem, may act in any number of different planes, so long as they all act through the point A.

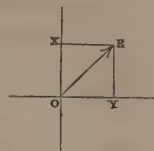
54. Resolution of Forces.—By reversing the process of the composition of forces, a single force may be resolved into two or more forces of which it is the resultant. These forces are called its *components*. Any given force may be resolved into components in an endless variety of ways. The following are the cases which will have to be referred to in this treatise:—

I. To resolve a given force into two components, acting along given lines through the same point and in the same plane. This is the converse of the proposition called the Parallelogram of Forces. Let A D (in Fig. 27) represent the given force; A B and A C the given lines in the same plane with A D, along which the components are

to act; through D draw D C and D B parallel respectively to the given lines, so as to make the parallelogram A B C D; A B and A C will represent the components required.

II. To resolve a given force into two rectangular components in a given plane. Let O R (Fig. 29) represent the given force, and through O let a pair of lines perpendicular to each other (called *axes*) be drawn, being the lines of action of the required rectangular components. Then from R draw R Y and R X parallel respectively to the two axes, so as to make the rectangle O X R Y; then O X and O Y will represent the required components.

Fig. 29.



This is evidently a particular case of the previous problem; but it has the following peculiarities, because of the triangle O X R being right-angled:—

The sum of the squares of the components is equal to the square of the resultant.

Each component is equal to the resultant multiplied by the cosine of the angle which the resultant makes with that component.

[In algebraical symbols, let R denote the resultant, X and Y the two rectangular components, α the angle between R and X, so that $90^\circ - \alpha$ is the angle between R and Y; then

$$R^2 = X^2 + Y^2;$$

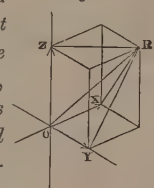
$$X = R \cos. \alpha; \quad Y = R \sin. \alpha.]$$

The resolution of the pressure of the wind on a ship's sails into a longitudinal and a transverse component, has already been mentioned in Chap. I., Art. 10.

III. To resolve a given force into three rectangular components.

Let O R (Fig. 30) represent the given force; and through O, let three lines, or *axes*, at right angles to each other, represent the lines along which the required components are to act. Then through the point R, let three planes pass, parallel respectively to the three planes which pass through the axes by pairs, so as to make a rectangular solid: the three edges of that rectangular solid, O X, O Y, O Z, will represent the three components required: or otherwise, from the point, R, let fall perpendiculars, R X, R Y, R Z, cutting the three axes; then O X, O Y, O Z, will be the components required.

Fig. 30.



In this case the sum of the squares of the three components is equal to the square of the resultant; and also, each component is equal to the resultant multiplied by the cosine of the angle which the resultant makes with that component.

[In algebraical symbols, let R denote the resultant, X, Y, and Z its three components, and α , β , and γ the three angles which they respectively make with it; then

$$R^2 = X^2 + Y^2 + Z^2;$$

$$X = R \cos. \alpha; \quad Y = R \cos. \beta; \quad Z = R \cos. \gamma.]$$

The application of the algebraical signs + and —, requires attention in computing and using rectangular components. To distinguish forces from each other which act in opposite directions along the same axis, those which act in one direction are to be treated as positive, and those which act in the contrary direction as negative. To fix the ideas, suppose for example that we have under consideration a force acting on a ship; that it is to be resolved into rectangular components acting along three axes, that of X longitudinal, that of Y transverse, and that of Z vertical.

Then if component forces acting $\left\{ \begin{array}{l} \text{forward} \\ \text{towards the right} \\ \text{upward} \end{array} \right\}$ be treated as positive, those acting $\left\{ \begin{array}{l} \text{backward} \\ \text{towards the left} \\ \text{downward} \end{array} \right\}$ must be treated as negative.]

55. *The Composition of Forces by the aid of Rectangular Components* is one of the chief uses of their resolution into such components. The following rule for finding the resultant of any combination of forces acting through one point, shows its principal application.

If all the forces act in one plane, two rectangular axes will be sufficient; if they act in different planes, three will be required.

Resolve each of the given forces into its rectangular components; take (by Rule I. of Art. 53) the resultant of the set of components acting along each of the axes separately. The resultants thus found may be called the resultant components; the resultant of those separate resultants (or resultant components) will be the complete resultant required. According to the principles already stated in Art. 54, the sum of the squares of the resultant components will give the square of the complete resultant; and the ratios of the resultant components to the complete resultant will be the cosines of the angles which it makes with the axes respectively. When a system of forces balance each other, each of the resultant components is separately = 0.

56. *Combination of a Single Force with a Couple.*—When a couple (Art. 35) tending to turn a body about in a given plane, is combined with a single force acting in a direction perpendicular to that plane, the combination cannot be further simplified.

It is otherwise when the force and the couple act in the same or in parallel planes; for then the combination is simply equivalent to a force equal and parallel to the single force, and in the same plane; but having its line of action shifted in a direction contrary to that of the couple, through a distance equal to the moment of the couple, divided by the magnitude of the single force.

For example, let AB (Fig. 31) represent a single force applied to a body in a given plane, and let C represent a couple applied to the same body, in the same or a parallel plane. Divide the moment of the couple by the single force; the quotient will be a certain distance, AE, which is to be laid off in the given plane at

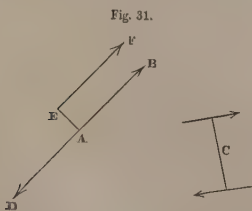
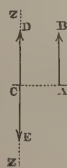


Fig. 32.



57. *Moment of a Force.*—It is sometimes requisite to reverse the process of the preceding article, by resolving a single force into another equal and parallel single force and a couple, according to the following principle:—*A given single force is equivalent to an equal and parallel single force acting in a different line, combined with a couple whose moment is the product of the magnitude of the force into the perpendicular distances between the lines of action, and whose direction is the same with that in which the new line of action lies relatively to the original force.*

For example, in Fig. 32, let AB represent the given force, and ZZ a new line of action parallel to AB, at the perpendicular distance, AC. Conceive that along the line, ZZ, there act the following forces: CD, equal to AB, and in the same direction, and CE, equal and opposite to AB. Then the original single force, AB, is equivalent to the new single force, CD, combined with the couple made up of the forces, AB and CE, with the arm AC; the moment of that couple is $AB \times AC$; and in the present example it is left-handed, the new line of action, ZZ, lying to the left of the direction of the original force, AB.

The moment of the couple thus found is called the *moment of the force*, AB, *relatively to the point*, C; or relatively to any other point in the line, ZZ; or relatively to any plane, or any axis, which traverses ZZ in a direction at right angles to the arm, AC.

58. *Balance of Couples.*—A couple can be balanced only by an equal and opposite couple; that is, by a couple of the same moment, acting in the same or in a parallel plane, and tending to turn the body in the opposite direction.

The resultant of a number of couples acting in the same plane, or in parallel planes, is a couple whose moment is the algebraical sum (see Art. 53) of their moments; that is to say, add together the moments of all the right-handed couples, and add together separately the moments of all the left-handed couples; the greater of those sums will indicate the direction of the resultant couple, whose moment will be equal to their difference. If that resultant moment is nothing, the couples are already balanced; if it has some value, then a couple of equal and opposite moment is what is wanted to make equilibrium.

There will be little occasion, in this treatise, to refer to the combination of couples acting in different planes. It may be sufficient, then, to state briefly, that if the moment of a couple be represented by the length of a line perpendicular to the plane in which the couple acts, and which is made to point in that direction in which an observer must look in order to make the couple seem right-handed, such a line is called the *axis of the couple*; and that by treating those axes exactly as the lines representing single forces are treated, couples may be compounded and resolved exactly like single forces.

59. *The Resultant of any number of Parallel Forces* is simply the algebraical sum of those forces; that is to say, it is of the same magnitude as if all the forces acted along one line, instead of acting along different parallel lines. (Art. 53, Rule I.)

The position of the line of action of the resultant of a number of parallel forces depends on the following principle:—*The moment of the resultant relatively to any given axis is equal to the resultant of the moments of the components.* Hence the following—

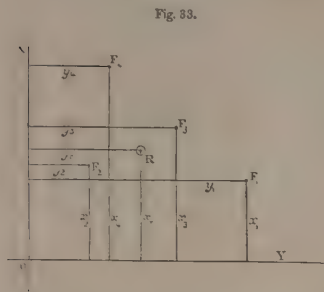
RULE.—*Having assumed any convenient axis in a plane perpendicular to the lines of action of the parallel forces, measure the perpendicular distances of the forces from that axis, and compute their moments relatively to it, by multiplying each force by its distance*

right-angles to AB, and in a direction contrary to that of the couple (that is to say, in the present example, towards the left of the force, AB, because the couple is right-handed). Then conceive that for the actual couple, C, there is substituted an *equivalent couple* (that is, a couple of the same moment, and acting in the same or in a parallel plane) made up as follows: a force, AD, equal and directly opposed to AB, and a force, EF, equal and parallel to AB, with the arm, AE. The forces AB and AD neutralize each other; and there remains, as the resultant of the single force and the couple, only the single force, EF, equal and parallel to the original force, but shifted through the distance, AE.

from the axis; distinguish those moments into right and left handed, and take their resultant, which divide by the Resultant Force; the quotient will be the perpendicular distance of that force from the assumed axis.

Find, by a similar process, the perpendicular distance of the Resultant Force from another axis at right angles to the first axis, and in the same plane; the position of the resultant will then be completely determined.

For example, in Fig. 33, let the parallel forces act perpendicular to the plane of the paper, and let F_1, F_2, F_3, F_4 , &c., be the points in that plane through which they act. Assume an axis, OY , and measure the perpendicular distance of each force from it; these are the distances marked x_1, x_2, x_3, x_4 . Multiply each force by its distance from the axis, OY ; the products will be the moments of the forces relatively to that axis; take the resultant of those moments,



distinguishing between right and left handed moments. Divide that resultant moment by the resultant of the forces; the quotient will be the distance of that resultant force from the axis, OY . Then take the distances, y_1 , &c., of the forces, from another axis, OX , perpendicular to OY , and proceed as before, to find the distance of the resultant from that new axis. The two distances from the axes, or *co-ordinates*, of the resultant, being thus known, the position of the point, R , through which it acts can be laid down.

[In algebraical symbols, let R denote the resultant force, and x_r, y_r its co-ordinates. Then

$$R = F_1 + F_2 + F_3 + \&c.;$$

$$x_r = \frac{F_1 x_1 + F_2 x_2 + F_3 x_3 + \&c.}{R};$$

$$y_r = \frac{F_1 y_1 + F_2 y_2 + F_3 y_3 + \&c.}{R}.$$

In these equations, all the terms have the sign +, on the supposition that all the forces act in one direction, and at the same side of each of the axes. When some of the forces act in one direction, and some in the opposite direction, they must be distinguished into positive and negative; and if they act some at one side of an axis and some at the other, their distances from that axis must also be distinguished into positive and negative; and in taking the products it is to be observed (agreeably to the rules of algebra) that a force and an ordinate of the same sign give a positive product, and of opposite signs a negative product.]

If the resultant moment relatively to an axis is nothing, it shows that the resultant force acts through a point in that axis. If the resultant force is nothing, and either or both of the resultant moments have some value, the resultant of the given combination of parallel forces is a couple, and not a single force.

If the resultant force and the two resultant moments are each of them nothing, the parallel forces are exactly balanced.

60. *The Resultant of any Combination of Forces, acting in any direction through any set of points*, is found by taking three rectangular axes, and resolving each force into components parallel to those axes, as in Art. 43. Thus are formed three sets of parallel forces, whose resultants are found separately and then combined.

61. The Centre of Gravity of a heavy body, or of a system of heavy bodies, is a point which is always traversed by the resultant of the weight of the body or system.

Supposing a set of weights to act at detached points, their common centre of gravity is found by the following—

RULE I.—Having chosen a fixed plane to which to refer the positions of the weights, take their moments relatively to that plane by multiplying each weight by its perpendicular distance from the plane; find the resultant of those moments, and divide it by the sum of the weights; the quotient will be the distance of their common centre of gravity from the fixed plane. By a similar process, find the distances of the same point from two other fixed planes at right angles to the first plane and to each other; the position of the centre of gravity will then be completely known.

The three planes are called, as in other cases, *co-ordinate planes*, and the distances of the weights and of their centre of gravity from those planes, *co-ordinates*.

The weights and their resultant (or sum) are to be regarded as all positive; but if some of the weights lie at one side, and some at the opposite side of a co-ordinate plane, their co-ordinates and moments relatively to that plane must be distinguished into positive and negative, as in previous examples of a similar kind. The sign of the resultant moment will show at which side of the plane the centre of gravity lies.

When the resultant moment relatively to a plane is nothing, the centre of gravity is in that plane; and when each of the resultant moments is nothing, the centre of gravity is at the point of intersection of the three planes, or *origin of co-ordinates*. In other words, the moment of a set of weights relatively to their common centre of gravity is nothing.

The two following particular cases are useful:—

II. *The common centre of gravity of two weights divides the straight line joining them into parts inversely proportional to the weights respectively furthest from them.* For example, in Fig. 34, let A

Fig. 34.



Fig. 35.

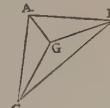
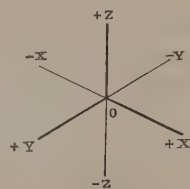


Fig. 36.



and B be the two weights; join AB ; the centre of gravity, G , divides that line in the following proportion:—

$$\frac{A+B}{AB} : \frac{A}{GB} : \frac{B}{GA}.$$

III. *The common centre of gravity of three weights is in the same plane with them; and if from the centre of gravity three straight lines be drawn to the three weights, those lines will divide the triangle formed by the weights into three triangles, each of an area proportional to the weight furthest from it.* In Fig. 35, let A, B, C , be the weights, G their common centre of gravity; then,

$$\frac{A+B+C}{\text{Triangle } ABC} : \frac{A}{\text{Triangle } GBC} : \frac{B}{\text{Triangle } GCA} : \frac{C}{\text{Triangle } GAB}.$$

[In algebraical symbols, the general problem of finding the common centre of gravity of a set of weights is expressed as follows:—In Fig. 36, let $-XO + X$, $-YO + Y$, $-ZO + Z$,

be the axes of co-ordinates, being the three straight lines in which the three co-ordinate planes cut each other. The distance or

ordinate of any point from the plane $\begin{Bmatrix} Y O Z \\ Z O X \\ X O Y \end{Bmatrix}$ is denoted by $\begin{Bmatrix} x \\ y \\ z \end{Bmatrix}$,

and regarded as positive or negative, according as the point lies to one side of the plane or to the opposite side. Let the weights be denoted by W_1, W_2, W_3 , &c.; their co-ordinates by $x_1, y_1, z_1; x_2, y_2, z_2; x_3, y_3, z_3$, &c.; and the co-ordinates of their common centre of gravity by x_0, y_0, z_0 ; then

$$\begin{aligned} x_0 &= \frac{W_1 x_1 + W_2 x_2 + W_3 x_3 + \&c.}{W_1 + W_2 + W_3 + \&c.}; \\ y_0 &= \frac{W_1 y_1 + W_2 y_2 + W_3 y_3 + \&c.}{W_1 + W_2 + W_3 + \&c.}; \\ z_0 &= \frac{W_1 z_1 + W_2 z_2 + W_3 z_3 + \&c.}{W_1 + W_2 + W_3 + \&c.} \end{aligned}$$

EXAMPLE.—Suppose that as the co-ordinate planes are taken, a horizontal plane in a ship's engine-room; a vertical longitudinal plane traversing that room amidships; and a transverse plane in a convenient assumed position; and that the weights under consideration are four portions of the engine, of the weights and in the positions stated in the following table:—

WEIGHTS.		CO-ORDINATES.					
No.	Lbs.	Longitudinal.		Transverse.		Vertical.	
		Forward.	Backward.	Right.	Left.	Up.	Down.
1	1000	2	...	3	...	5	...
2	500	...	1	...	3	5	...
3	250	2	...	3	2
4	500	...	3	...	6	0	0
Total Weight,...		2250					

Then the following table shows the calculation of the moments and resultant moments, and of the co-ordinates of the centre of gravity:—

MOMENTS RELATIVELY TO							
No. of Weight.	Transverse Plane.		Longitudinal Plane.		Horizontal Plane.		
	Positive.	Negative.	Positive.	Negative.	Positive.	Negative.	
1	2000	...	3000	...	5000	...	
2	...	500	...	1500	2500	...	
3	500	...	750	500	
4	...	1500	...	3000	0	0	
Sums,.....	2500	2000	3750	4500	7500	500	
Subtract,...	2000	3750	500	...	
Resultant Moments,}	500	750	7000	...	

Divide by total weight, 2250, giving for the co-ordinates of the centre of gravity,.....} Forward, 0.22 | Left, 0.33 | Up, 8.11

IV. To find the effect upon the position of the centre of gravity of a set of weights, of shifting one of those weights into a new position, multiply the weight shifted by the distance through which it is shifted, and divide by the sum of all the weights: the quotient will be the distance through which the centre of gravity will be shifted, in a direction parallel to that in which the weight is shifted.

V. To find how far a given single weight must be shifted, in order to shift the common centre of gravity through a given distance in the same direction, multiply the sum of the weights by the distance through which their common centre of gravity is to be shifted, and divide by the single weight.

For example, if weight No. 3 of the preceding table, 250 lbs., be shifted in any direction through $4\frac{1}{2}$ feet, the centre of gravity will be shifted in the same direction through

$$\frac{250 \times 4\frac{1}{2}}{2250} = \frac{1}{2} \text{ foot.}$$

62. *Centres of Gravity and Moments of Bodies.*—The supposition of the weight of a body being concentrated at a point is a mathematical fiction, as has been already stated in Chap. I., Art. 3; but the centre of gravity of a body, at which its weight may be conceived to be concentrated, without error as regards its mechanical action as a whole, can always be found.

When the body is *homogeneous*, or composed of material of uniform heaviness throughout, the following principles serve to determine the centre of gravity and the moment of its weight relatively to a given plane.

I. *The centre of gravity of a homogeneous body is its centre of figure, or of mean distances.*

II. *The moment of the weight, or statical moment, of a homogeneous body, relatively to a given plane, is equal to the product of its geometrical moment relatively to that plane into the heaviness of the material.*

Thus all the rules given in Section II. of this chapter, Articles 32 to 42 inclusive, for finding the centres and moments of figures, together with the examples of those rules, become at once applicable to the purpose of finding the centres of gravity and statical moments of homogeneous bodies of those figures.

The rules which have reference to *plane areas* are applicable to plates or prismatic bodies of uniform thickness, having those areas for bases. In computing statical moments by means of them, the following rule is to be used:—

III. *Multiply the geometrical moment of the plane base by the thickness, and by the heaviness of the material.* The product of the thickness into the heaviness is the weight per unit of area.

In all such calculations, if the dimensions are expressed in feet, and the heaviness in lbs. to the cubic foot, the moment is expressed in *foot-pounds*. If the heaviness is that of *sea-water*, the geometrical moment itself is the statical moment in *cubic feet of sea-water at a leverage of a foot*; which may be multiplied by 64 for *foot-pounds*, or divided by 35 for *foot-tons*.

When a body is *heterogeneous*, or consists of parts of different heaviness, the following rule is to be applied:—

IV. *Divide the body into parts, each of which is of uniform or sensibly uniform heaviness; find the centre of gravity of each such part separately; conceive the weight of each part to be concentrated at its own centre of gravity, and treat those weights as detached weights* (according to the rules of Art. 61).

63. *The Resultant of a Pressure distributed over a plane surface* is found by the following rules:—

I. If the intensity (see Art. 51) is uniform, multiply the area of the surface by the intensity.

II. If the intensity is not uniform, conceive that the surface lies horizontal, and that a solid stands upon it, whose height at each point represents the intensity of the pressure at that point. Then the volume of that solid will represent the amount of the pressure. As to finding the volume of such a solid, see Section I. of this chapter, Articles 24, 26, 28.

The *Centre of Pressure* means, a point traversed by the resultant of a pressure that is distributed over a surface. When the surface pressed upon is plane, the centre of pressure is a point in that surface itself, and is found according to the following rules:—

III. When the intensity of the pressure is uniform, the centre of figure is the centre of pressure.

IV. When the intensity of the pressure is not uniform, find the centre of the solid of Rule II., from which let fall a perpendicular

on the pressed surface; the foot of that perpendicular will be the centre of pressure; or otherwise, find the co-ordinates of the centre of that solid relatively to two axes in the plane of the pressed surface: these will be the co-ordinates of the centre of pressure. (See Art. 38.)

V. If the pressure is that of a fluid upon a solid body immersed on it, then, as already stated in Chapter I., Articles 2 and 3, the Resultant Pressure is equal and opposite to the weight of the volume of fluid displaced; and the centre of pressure is the centre of that volume, or CENTRE OF DISPLACEMENT.

64. *Uniform Motion under Balanced Forces.—Action of Machines.*—A body, or system of bodies, which is under the action of no force, or of balanced forces, moves uniformly

The relations amongst the forces which act on a body, or a system of bodies, such as a machine, moving uniformly, are expressed in a convenient form for many practical purposes, by saying that *in any given time the energy exerted is equal to the work performed*. The terms “energy” and “work” have been, to a certain extent, explained in Chap. I., Art. 9, and in the present Section, Art. 47; the following are some additional explanations:—

A force applied to a moving point, in the direction of its motion, is a *driving* or *propelling* force, or an *effort*. If an effort be multiplied by the distance through which it drives the point that it acts upon, the product is called the *energy exerted* by that effort

A force applied to a moving point in a direction opposite to that of its motion, is an *opposing* force, or *resistance*. If a resistance be multiplied by the distance through which the point that it acts upon is driven against it, the product is called the *work performed* in overcoming that resistance.

A force applied to a moving point in a direction at right angles to that of its motion, is a *lateral* force; and it does not directly affect the energy exerted nor the work performed.

In Fig. 37, let A represent a point in a body, or in a machine, which point is in the act of moving in the direction AB with a given velocity. Let AF represent a force applied to the point A,

Fig. 37.

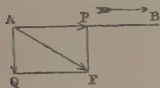
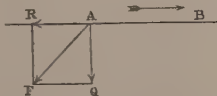


Fig. 38.



in a direction obliquely forwards. From F let fall FP perpendicular to AB, and complete the rectangle, APFQ. The force, AF, is thus resolved into two—an *effort*, AP, and a lateral force, AQ; and the effort alone is to be considered in calculating the *energy exerted*.

In Fig. 38, let A represent a point in a body, or in a machine, which point is in the act of moving in the direction AB with a given velocity. Let AF represent a force applied to the point A, in a direction obliquely backwards. From F let fall a perpendicular FR on BA produced, and complete the rectangle ARFQ. The force, AF, is thus resolved into two—a *resistance*, AR, and a lateral force, AQ; and the resistance alone is to be considered in calculating the *work performed*.

The application of the principle of the equality of energy exerted to work performed leads to the following rules:—

I. To find the energy which must be exerted in order to make a given body or machine perform a given motion against given resistances at an uniform speed: *Multiply each resistance by the distance through which it is to be overcome; add together the products;*

the sum will be the total work to be performed, which will be equal to the energy to be exerted. (For an elementary illustration, see back, Chap. I., Article 9.)

II. To find the effort or propelling force required, in order to drive a given body or machine at an uniform speed against given resistances: *Divide the energy to be exerted in a given time by the distance to be moved through by the point of application of the effort (or driving point, as it is called); or otherwise,*

III. *Multiply each resistance by the ratio which the velocity of the point where it is overcome bears to the velocity of the driving point; the sum of the products will be the required effort.* This last process of calculation is called the “reduction of resistances to the driving point;” and the principle on which it depends, the “principle of virtual velocities;” because the result does not depend on the absolute or actual velocities of the points where the forces are applied, but only on the proportions which those velocities bear, or might bear, to each other.

65. *Useful and Wasteful Work.—Efficiency of a Machine.*—As already stated in Chap. I., Art. 9, the work performed by a machine is distinguished into *useful work*, being that for the purpose of performing which the machine is planned, and *wasteful work*, being all the other work which the machine performs in overcoming resistances which are of no service towards the performance of the useful work, such as the friction of its own mechanism.

The proportion which the useful work bears to the whole work, wasteful as well as useful, is called the *efficiency* of the machine. The efficiency of a perfect machine, were such a machine possible, would be represented by unity; the efficiency of every actual machine is represented by a fraction, which approaches nearer to unity as the machine approaches nearer to perfection.

EXAMPLE.—Required, the efficiency of the mechanism and paddle-wheels of a steam-ship, under the following circumstances:—The speed of the ship being 12 knots per hour, or $20\frac{1}{2}$ feet per second, the resistance of the water to the motion of the ship was 12900 lbs. At the same time, the mean speed of the pistons was 3·4 feet per second, and the amount of the effort, being the total mean effective pressure of the steam on them, was 120875 lbs.

$$\begin{aligned} \text{Useful Resistance, } 12900 \times \text{Speed, } 20\frac{1}{2} &= 261225 \text{ foot-lbs.,} \\ &\text{being the useful work performed per second} \\ \text{Effort, } 120875 \times \text{Speed of driving point, } 3\cdot4 &= 410975, \\ &\text{being the energy exerted, or total work performed per second;} \\ \text{Useful work, } 261225 & \\ \text{Total work, } 410975 &= \cdot 633 \text{ efficiency.} \end{aligned}$$

This shows that about 63 per cent. of the whole work performed by the steam was usefully performed in driving the vessel, the remaining 37 per cent. being expended in overcoming the friction of the engine, driving the water backwards, &c.

66. The term *Mechanical Power* has often been used in the same sense with “energy;” that is, the means of performing, or capacity to perform, a certain quantity of work, without reference to the time occupied in performing that work; but strictly speaking, it is more correct to use the term *Power* to denote a supply of energy at a certain rate in a certain time; such as so many foot-pounds per second, per minute, or per hour.

The terms *One Horse-power*, or *Real Horse-power* (in contradistinction to “nominal horse-power,” which will be afterwards explained) denotes a supply of energy at the rate of—

$$\begin{aligned} &550 \text{ foot-pounds per second, or} \\ &33,000 \text{ “ “ per minute, or} \\ &1,980,000 \text{ “ “ per hour.} \end{aligned}$$

Thus the engines of the steam-ship referred to in the last article, which exerted a total energy of 410975 foot-pounds per second, were of—

$$\frac{410975}{550} = 747 \text{ horse-power nearly.}$$

The mutual actions amongst the parts of a body or machine are not in any way affected by the body or machine being transported, as a whole in any direction, with any uniform motion, how swift soever. For example, the actions of the parts of a marine steam-engine on each other are in no way affected by the fact of the whole engine being transported bodily along with the vessel, nor by the fact of the vessel and the sea in which she floats being transported along with the earth. In short, the actions of bodies, or of the parts of a body, on each other, are affected by their motions relatively to each other only.

67. *Energy and Work of Couples.*—When a body turns against a resisting couple, being driven round by a propelling couple, the energy exerted by the driving couple, and the work performed against the resisting couple, may be computed in the ordinary way by multiplying each force by the distance moved through by its point of application with or against the force, as the case may be. But it is sometimes more convenient to obtain the same result by multiplying the moment of the couple (Art. 46) by the turning motion of the body in circular measure (Arts. 30, 49).

For example, suppose that the paddle-wheel of a steam-vessel makes 24 turns per minute; that its effective radius is 10.25 feet (so that its circumference is 64.4028 feet); and that it overcomes a resistance of 6450 lbs. This is the force exerted between the water and the paddle-floats; an equal and opposite force is exerted between the paddle-shaft and its bearings, making a couple with a force of 6450 lbs., and an arm of 10.25 feet.

Then, according to the ordinary way of computation, we have—

Resistance overcome,.....	6450 lbs.
Distance moved in a minute by its point of application, } 24 × 64.4028 =	1545.6672 feet.

Work performed per minute,..... 9,969,553 ft.-lbs.

Calculating by the moment and angular motion, we have—

Moment of resisting couple, 6450 × 10.25 =	66112.5
Circular measure of motion in a minute, 24 × 6.2832 =	150.7968

Work performed per minute (as before),..... 9,969,553 ft.-lbs.

68. *Periodical Motion.*—When a body or a machine is in a state of periodical motion; that is to say, when its motion, although undergoing temporary fluctuations, returns at the end of certain periods of time to a certain fixed speed—the principles and rules of the three preceding articles, though not applicable to the motion at every instant, are applicable to the motion during each complete period. That is to say, the energy exerted by the driving force during the complete period is equal to the total work performed against resistances during the same period; and the relations amongst the resistances and driving forces, and the velocities or virtual velocities of their points of application, which hold at every instant during uniform motion, are true of the mean values of those quantities during periodical motion. For example, in an ordinary steam-engine, there are many parts whose velocities are continually varying—the piston itself, to which the driving force is applied, being one of them; but at the end of each revolution each part returns to its original velocity; and accordingly, in each complete revolution, the energy exerted is equal to the total work performed,

and the mean effort exerted by the steam on the piston is such as exactly to balance the mean total resistance. Thus, when a body or a machine is in a state of periodical motion, many mechanical questions respecting it can be solved without taking the fluctuations of speed into account.

69. *Representation of Energy and Work by Areas.*—The energy exerted by a constant effort, or the work performed against a constant resistance, may be represented by the area of a rectangle, of which the base represents the distance moved through by the point of application of the force, and the height represents the force.

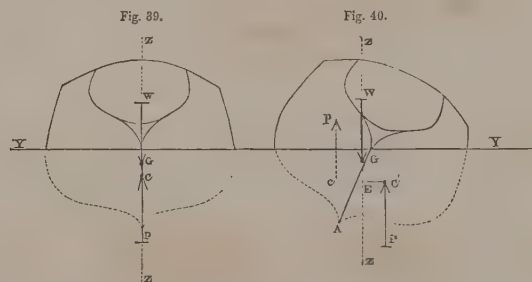
When the force varies in amount, let abscissæ measured along the base of a plane figure be conceived to represent distances moved through, while the ordinates represent the corresponding amounts of the effort or resistance, as the case may be (see Art. 19). Then the area of the figure, found by means of a suitable rule, will represent the energy or work, as the case may be. The mean breadth of the figure (see Art. 20) will represent the mean value of the force.

Figures of this kind, called *cards* or *diagrams*, drawn by instruments called *indicators*, or *self-registering dynamometers*, are used in ascertaining the power of steam-engines and other machines, and will be further explained in the sequel. The areas of such diagrams may be measured by means of the Platometer (Art. 23).

70. *Stable and Unstable Equilibrium.*—The meaning of those terms, especially in their application to ships, has already been sufficiently explained in Chap. I., Article 4.

It now only remains to explain the terms of *statical* and *dynamical stability*; and for that purpose, Figs. 5 and 6 of Chap. I. are here repeated as Figs. 39 and 40.

The *Statical Stability* of a body is measured, and expressed as a quantity, either by the righting force (or righting couple, as the



case may be) corresponding to some given deviation of position, or by the proportion which the righting force or couple bears to the corresponding deviation.

The statical stability of the ship in Figs. 39 and 40 may be expressed by saying, that for the “heel” or angular deviation, ZGA, the moment of the righting couple is = weight of the ship × EC', or by stating the proportion—

$$\text{Sine of angle } \frac{W \times EC'}{ZGA}$$

Dynamical Stability (a term introduced by the Reverend Canon Moseley) means the quantity of mechanical work done in causing a body to deviate to a certain extent from its position of equilibrium. It is stated in foot-pounds, and may be found, if the deviation consists in bodily shifting, by multiplying the total deviation by the mean righting force; and if the deviation consists in a change of angular position, by multiplying the total deviation, in circular

measure, by the mean moment of the righting couple. A simpler method of computing the dynamical stability is the following: multiply the weight of the body by the increase of elevation which its centre of gravity undergoes relatively to the point of application of the supporting force. For example, in Fig. 40, where the centre of buoyancy, C', is to be held as the point of application of the supporting force, the dynamical stability is—

$$W \times (GE - GC);$$

for GC, in Fig. 39, is the height of the centre of gravity above the centre of buoyancy in the position of equilibrium, and GE, in Fig. 40, is the corresponding height in the deviated position.

The latter method is connected with the proposition, that *when a body is stable, the position of its centre of gravity relatively to its point of support is the lowest possible.*

71. Effects of Unbalanced Forces.—An unbalanced force produces motion in the body on which it acts; and that motion is combined with the motion which the body already has. For such purposes as those of the present treatise, the most convenient mode of classing the unbalanced forces which may act on bodies is the same with that already adopted in Art. 53 for balanced forces; that is to say, Efforts, or forces which act in the direction of the body's motion; Resistances, or forces which act against the direction of the body's motion; and Lateral Forces, which act at right angles to the direction of the body's motion.

An unbalanced effort increases the speed of the body's motion, and is called an *accelerating force*; an unbalanced resistance diminishes the speed of the body's motion, and is called a *retarding force*; an unbalanced lateral force changes the direction of the body's motion, causing it to move in a curve, and is called a *deviating force*.

The law according to which an unbalanced force produces motion in a body, may be expressed in the following form: the *momentum produced is equal to the impulse exerted.*

Impulse means the product of a force into the time during which it acts; the force, in British measures, being usually expressed in pounds, and the time in seconds. A small force acting for a long time may exert as great an impulse as a great force acting for a short time. Impulses are compounded and resolved like forces.

Momentum means the product of the mass of a body into its velocity. The mass of a body is proportional to the force which is required to impress a given velocity on the body in a given time. It is known that the gravitation of bodies towards the earth acting for equal times, impresses equal velocities on all substances whatsoever, that are similarly situated with respect to the earth's surface; hence the masses of bodies are proportional to their weights in a given locality. To find the mass of a body in units suitable for computing its momentum, divide the body's weight by the velocity which that weight produces in the body during one second of unresisted fall. The mean value of that velocity, in feet per second, is about 32.2. It is called, the *accelerating effect of gravity*, and in algebraical formulæ is usually denoted by the letter *g*.

The strict meaning of the word *density* is, the mass of an unit of volume; hence, to find the density of a body in British measure, divide its heaviness by 32.2. The word "density" is seldom used with the precision implied in this definition, being employed sometimes in the sense of specific gravity, and sometimes in that of heaviness (see Art. 50).

72. The Effect of an Uniform Accelerating Force, or unbalanced effort, on the motion of a body, depends on the following principles:—

The force multiplied by the time during which it acts, gives the impulse exerted by it; and this is equal to the increase of momentum; that is, the product of the mass of the body, by the increase of velocity which the body undergoes between the beginning and end of the action of the accelerating force.

Multiply both those equal quantities—that is to say, the impulse, and the momentum which it produces—by the mean velocity of the body; the products will be equal. Consequently, the product of the force into the time into the mean velocity, or in other words, of the force into the distance through which it acts,—that is, the energy exerted by the force, will be equal to the product of the mass, into the increase of the velocity, into the mean velocity; that is, to the product of the mass into the increase in the half square of the velocity.

These principles are summed up in the two following sets of equations, by means of which any problem respecting uniform acceleration may be solved:—

$$\text{I. IMPULSE} = \text{Force} \times \text{Time} = \text{Mass} \times \text{Increase of Velocity} \\ = \frac{\text{Weight} \times \text{Increase of Velocity}}{\text{Gravity (32.2)}} = \text{INCREASE OF MOMENTUM.}$$

$$\text{II. ENERGY exerted} = \text{Force} \times \text{Distance} = \text{Force} \times \text{Time} \times \text{Mean Velocity} \\ = \text{Mass} \times \text{Increase of half-square of Velocity} \\ = \frac{\text{Weight} \times \text{Increase of square of Velocity}}{2 \times \text{Gravity (64.4)}}.$$

[In algebraical symbols, let P be the unbalanced effort; *t*, the time during which it acts, in seconds; *s*, the distance through which it acts, in feet; W, the weight of the body; *v*₀, the original velocity; *v*₁, the increased velocity; then—

$$\text{I. } P t = \frac{W (v_1 - v_0)}{g};$$

$$\text{II. } P s = \frac{P t (v_1 + v_0)}{2} = \frac{W (v_1^2 - v_0^2)}{2g}.]$$

EXAMPLE.—A certain ship, together with the mass of water which she sets in motion with her own speed, weighs 1000 tons = 2,240,000 lbs. Her speed, being 16.875 feet per second, is to be increased to 20.25 feet per second.

Question I. What effort (over and above that required to overcome the resistance) must be exerted for 60 seconds, in order to produce that acceleration?

Weight of body,.....	2,240,000
× Acceleration, 20.25—16.875,.....	3.375
Divide by gravity,..... 32.2)	7,560,000 product.
Increase of Momentum = Impulse required =	234783
Impulse, 234783	
Time, 60	= 3913 lbs., effort required.

Question II. What effort (over and above that required to overcome the resistance) must be exerted through a distance of 1113.75 feet, in order to produce the same acceleration?

Square of Final Velocity,.....	410.062500
Less Square of Original Velocity,.....	284.765625
	2) 125.296875
Difference of Half-squares,.....	62.6484375
× Weight of body,.....	2,240,000
Divide by gravity, 32.2)	140,332,500
Energy to be exerted,.....	4,358,152 ft.-lbs.
Energy, 4,358,152	
Distance, 1113.75	= 3913 lbs. effort required.

The effort proves to be the same as before; the reason being, that the distance 1113·75 feet was purposely chosen, being that which is described in 60 seconds with the mean velocity of—

$$\frac{20\cdot25 + 16\cdot875}{2} = 18\cdot5625 \text{ feet per second.}$$

73. *The Effect of an Uniform Retarding Force* is determined by the same rules with the effect of an uniform accelerating force, with the following modifications:—

For *increase* in the velocity, the momentum, or the half-square of the velocity, is to be substituted *diminution*; for *acceleration*, *retardation*; and for *energy exerted*, *work performed*. The reason for the last of those changes is, that in order to diminish a body's speed, a resistance must be opposed to its motion; during the retardation the body overcomes that resistance, and so performs work measured by the product of the resistance into the distance through which it is overcome; and that work is exactly equal to the energy which would be required to restore the velocity from its diminished value to its original value.

[To adapt the algebraical formulæ to the case of retardation, it is only necessary to put $v_0 - v_1$ instead of $v_1 - v_0$, $v_0^2 - v_1^2$ instead of $v_1^2 - v_0^2$, R (denoting the unbalanced resistance) instead of P (denoting an unbalanced effort), and Rs (denoting work performed), instead of Ps (denoting energy exerted).

The computation given as an example may also be made to apply to a case of retardation, by supposing the question to be as follows:—What must be the excess of the resistance over the driving effort, in order that the speed of the ship may diminish from 20·25 feet per second to 16·875 feet per second, in 60 seconds (or in a distance of 1113·75 feet, as the case may be)?

74. *The Free Action of Gravity* on a body let fall from a state of rest, is one of the particular cases embraced by the rules of Art. 72; and those rules are made applicable to it by supposing the unbalanced effort to be simply the body's own weight. The following are the results:—

I. A body falls vertically downwards from a state of rest: to find the velocity, in feet per second, acquired at the end of a given time—*multiply the time in seconds by 32·2*.

II. To find the depth of fall in feet—*multiply the time by half the velocity acquired at the end of it; or otherwise, multiply the square of the time in seconds by 16·1*.

III. To find the time occupied in falling a given depth from a state of rest—*divide the depth of fall by 16·1, and extract the square root of the quotient*.

IV. To find the velocity, in feet per second, acquired in falling vertically from a given height—*multiply the height in feet by 64·4, and extract the square root of the product; or otherwise, multiply the square root of the height by 8·025*.

V. To find from what height in feet a body must fall in order to acquire a given velocity in feet per second—*divide the square of the velocity by 64·4*.

The velocity acquired by a body in falling from a given height is called "*the velocity due to that height*;" and the height from which a body must fall in order to acquire a given velocity, is called "*the height due to that velocity*."

[In algebraical symbols, let t denote the time; v , the velocity acquired at the end of that time; g , the accelerating effect of gravity in an unit of time; h , the height or depth of fall. Then—

$$\text{I. } v = gt; \quad \text{II. } h = \frac{vt}{2} = \frac{gt^2}{2}; \quad \text{III. } t = \sqrt{\frac{2h}{g}};$$

$$\text{IV. } v = \sqrt{2gh}; \quad \text{V. } h = \frac{v^2}{2g}.]$$

75. By the *Energy* or *Actual Energy* of a moving body is meant, the whole energy which must have been exerted on that body to bring it from a state of rest to its actual speed; being also the whole work which the moving body is capable of performing against a retarding resistance before being brought to a state of rest. To find the actual energy of a body of a given weight moving with a given speed—*multiply its weight by the height due to its velocity; or otherwise, multiply the weight in pounds by the square of the velocity in feet per second, and divide by 64·4*.

$$[\text{In algebraical symbols, } \frac{Wv^2}{2g}.]$$

When a body or machine moves with a fluctuating speed, acceleration is produced by an excess, and retardation by a deficiency of energy exerted as compared with work performed. During acceleration, the whole excess of energy is employed in increasing the actual energy of the motion; and during retardation, the deficiency of energy is made up at the expense of the actual energy of the motion; so that the actual energy of the body or machine may be regarded as a *store*, which receives the excess of the energy of the driving force at one time, and gives out the energy so received to make up for a deficiency at another time.

The term *actual* is applied to the energy of a moving mass, to distinguish it from that of a driving force capable of acting through a given distance, which is called *potential* energy.

The term "*vis viva*," or "*living force*," might be used in the same sense with actual energy; but as most writers on mechanics use it to denote the *product of the mass into the square of the velocity*, which is the double of what is denoted by actual energy, its employment is inconvenient, and apt to lead to mistakes.

76. *Reaction of Accelerated and Retarded Bodies*.—The principle of the equality of Action and Reaction, stated in Article 52, when applied to bodies undergoing acceleration or retardation, shows that *every mass undergoing acceleration reacts backwards, with a force equal and opposite to the effort producing the acceleration*; and that *every mass undergoing retardation reacts forwards with a force equal and opposite to the resistance producing the retardation*.

The applications of this principle are numerous and important. In Article 9 of Chapter I., it has already been stated how the reaction of the water is made use of to propel a vessel. This will be described in detail in a later chapter; and it will also be shown, in the division that relates to steam machinery, how the reactions of the moving parts of an engine are to be balanced against each other, so that they may not overstrain the mechanism and framework.

77. *Rotation Accelerated and Retarded*.—*Moment of Inertia*.—When a body of a given weight whirls in a circle round an axis at the end of an arm of a given length, the force required to produce a given acceleration or retardation of its speed may be computed by the rules of Articles 72 and 73. But in many cases it is convenient to use a method specially adapted to circular motion, which is as follows:—

Instead of the Effort or Resistance, use the *Moment* of that Effort or Resistance—viz., the product of the Effort by its leverage, or perpendicular distance from the axis.

Instead of the Velocity, use the *Angular Velocity* in units of circular measure per second (Art. 49), (which is equal to the velocity divided by the arm, or to the number of turns per second $\times 6.2832$).

Instead of the Weight of the body, use the *Moment of Inertia of its Weight*, which is the product of the weight into the square of the arm.

Then the rules of Articles 72 and 73 are transformed into the following:—For Acceleration—

I. Moment of Effort \times Time

$$= \frac{\text{Moment of Inertia of Weight} \times \text{Increase of Angular Velocity}}{\text{Gravity (32.2)}}$$

II. Energy exerted = Moment of Effort \times Circular motion during its action

= Moment of Effort \times Time \times Mean angular velocity

$$= \frac{\text{Moment of Inertia of Weight} \times \text{Increase of square of angular velocity}}{2 \times \text{Gravity (64.4)}}$$

[In algebraical symbols, let M denote the moment of the effort in foot-pounds; t , the time during which it acts, in seconds; W , the weight of the body in pounds; r , the length of the arm at the end of which it revolves, in feet; a_0 , its original angular velocity; and a_1 , its increased angular velocity; θ , the circular motion during the action of the effort, in circular measure—

$$\text{I. } M t = \frac{W r^2 (a_1 - a_0)}{g};$$

$$\text{II. } M \theta = M t \frac{a_1 + a_0}{2} = \frac{W r^2 (a_1^2 - a_0^2)}{2g}.$$

These formulæ take the following shape, when the speed of the circular motion is expressed in turns per second. Let n_0 be the original, and n_1 the increased number of turns per second; $2\pi = 6.2832$, the ratio of the circumference of a circle to its radius; then—

$$\text{I.}^\circ M t = \frac{2\pi W r^2 (n_1 - n_0)}{g};$$

$$\text{II.}^\circ M \theta = \frac{4\pi^2 W r^2 (n_1^2 - n_0^2)}{2g}.$$

The same rules are made applicable to retarded circular motion, by putting Resistance for Effort, Diminution for Increase of the velocity and of its square, and Work performed for Energy exerted.

The chief use of this method is when a system of bodies, carried by arms of different lengths, or a continuous body of any figure and dimensions, turns about one axis with one angular velocity; for then, to solve any question of acceleration or retardation, it is only necessary to take the sum of the moments of inertia of the weights of the system of bodies, or of the particles of the continuous body, and treat that total moment of inertia precisely as the moment of inertia of a single weight is treated in the preceding rules.

III. To find the *Actual Energy* of a body rotating about an axis, multiply the moment of inertia of its weight by the half-square of its angular velocity, and divide by 32.2.

78. *Moments of Inertia and Radii of Gyration*.—The radius of gyration of a solitary heavy particle, is the arm at the end of which it revolves round an axis; and the same is the case with a circular ring described about the axis. For a system of bodies, or a continuous body of any size, the square of the radius of gyration is found by dividing the total moment of inertia of the weight by the total weight.

In Articles 43 and 44 are given various rules and results as to the moments of inertia of plane figures. All these are made

applicable to thin flat plates of the same form, by the following rule:—

I. Multiply the moment of inertia of the plane figure by the weight of unity of area. A rule is also given as to radii of gyration about parallel axes, which, when applied to the weights of bodies, is as follows:—

II. The square of the radius of gyration of a body about an axis not traversing its centre of gravity, is equal to the square of its radius of gyration about a parallel axis traversing its centre of gravity, added to the square of the distance between the two axes. This rule has an important practical application in the stowage of ships, which will be explained further on.

III. The squares of the radii of gyration of similar figures are as the squares of their corresponding dimensions; and if those figures consist of material of equal heaviness, their moments of inertia are as the fifth powers of their corresponding dimensions.

As the Table in Article 44 is confined to plane figures, the following Table is added, containing the squares of the radii of gyration of certain solid figures:—

FIGURE.	AXIS.	SQUARE OF RADIUS OF GYRATION.
Sphere,.....	Any Diameter,.....	$\frac{2 \times \text{radius}^2}{5}$
Spheroid,.....	Polar Axis,.....	$\frac{2 \times \text{equatorial radius}^2}{5}$
Cylinder,.....	Longitudinal Axis,.....	$\frac{\text{Radius}^2}{2}$
Hollow Sphere,.....	Any Diameter,.....	$\frac{2 \times \text{Difference of 5th powers of radii}}{5 \times \text{Difference of cubes of radii}}$
Hollow Cylinder,.....	Longitudinal Axis,.....	$\frac{\text{Sum of Squares of radii}}{2}$
Rectangular Block,...	Longitudinal Axis,.....	$\frac{\text{Breadth}^2 + \text{thickness}^2}{12}$
Rectangular Block,...	Longitudinal Edge,....	$\frac{\text{Breadth}^2 + \text{thickness}^2}{3}$

79. *Impulse on a Free Solid Body*.—If a solid body is free to move in any manner, the effects of impulses applied to it are as follows:—

I. A single impulse, acting through the centre of gravity of the body, impresses on the body a motion of *translation* or *shifting*, in the direction of the impulse; that is to say, all the particles of the body move in parallel directions with the same speed, and the body as a whole shifts its position without turning; and the velocity of translation is—

$$\frac{\text{Force} \times \text{time} \times \text{gravity (32.2)}}{\text{Weight of body}}$$

II. A pair of equal and opposite impulses, acting in parallel lines—in other words, the *impulse of a couple*—impresses *no motion* on the centre of gravity of the body: on the body as a whole, it impresses a motion of *rotation* or *turning* about its centre of gravity; and the angular velocity, in circular measure, is—

$$\begin{aligned} & \frac{\text{Moment of Couple} \times \text{time} \times \text{gravity}}{\text{Moment of Inertia of Body's Weight}} \\ &= \frac{\text{Force} \times \text{arm} \times \text{time} \times \text{gravity}}{\text{Weight} \times \text{square of radius of gyration}} \end{aligned}$$

This is illustrated by the action of a pair of screw propellers on parallel shafts, working with equal power, one ahead and the other astern, which make the vessel turn about her centre of gravity without shifting her position; and a similar effect is produced by “filling” one set of sails and “backing” another.

A comparison of those effects, of a force and of a couple respectively, leads to the following rule, for finding the angular velocity produced by the impulse of a couple, when the linear velocity that would be produced by one of the two single impulses is known.

Angular velocity

$$= \frac{\text{Velocity produced by single impulse} \times \text{arm of couple}}{\text{Square of radius of gyration}}$$

III. When a single impulse acts in a line not traversing the centre of gravity, it is to be resolved, according to the principles of Article 57, into an equal and parallel single impulse acting through the centre of gravity, and the impulse of a couple whose force is the same with that of the single impulse, and whose arm is the perpendicular distance of the line of action of the actual single impulse from the centre of gravity; and the motion impressed on the body will be compounded of the translation due to the single impulse, and the rotation about the centre of gravity due to the impulse of the couple.

In Fig. 41, let G be the centre of gravity of the body (for example, a ship, seen in plan), C, the point of application of an impulse (for example, the rudder), and CP, the force producing the impulse, acting in a line which passes at the distance, GB, from the centre of gravity (GB being perpendicular to CP). The actual impulse is equivalent to an equal and parallel single impulse applied to the centre of gravity G, and producing translation in the direction indicated by the arrow v , with the velocity given by Rule I. of this article, combined with the impulse of a couple of the force CP, and arm GB, producing rotation in the direction indicated by the arrow a , with the angular velocity given by Rule II.

But the combined translation and rotation are equivalent to one movement, which is determined as follows. Draw GD perpendicular to BG, to represent the *radius of gyration* of the body; join BD; draw DA perpendicular to BD cutting BG produced in A; then the combined translation and rotation are equivalent simply to a rotation, in the same direction, with the same angular velocity, about an axis traversing A instead of G (called the *Instantaneous Axis*).

The distance GA of the instantaneous axis from the centre of gravity, as found by calculation, is as follows:—

$$GA = \frac{\text{Velocity of translation}}{\text{Angular velocity}} = \frac{\text{Square of radius of gyration}}{\text{Arm, GB}}$$

The direction of the motion impressed on any point in the body, such as E, is at right angles to EA.

These principles have important applications to the manoeuvring qualities of ships, as will be afterwards explained.

80. *Deviating Force*.—An unbalanced lateral force, as already stated in Article 71, compels the body to which it is applied to move in a curve, and is called a *deviating* or *deflecting force*. When the body revolves in a circle with an uniform speed, the deviating force must be directed towards the centre of that circle; and its amount is computed as follows:—

I. *Multiply the weight of the body by the square of its velocity in feet per second; divide by gravity (32.2) and by the radius of the circle in feet; the quotient will be the deviating force required.*

[In symbols, let W be the weight, v its velocity, r the radius of the circle, Q the deviating force; then

$$Q = \frac{Wv^2}{gr}.]$$

It is sometimes more convenient to express the speed of revolution by means of revolutions per second, or by means of angular velocity in circular measure per second; and then (observing that angular velocity = revolutions per second $\times 6.2832$, = linear velocity \div radius) the rule takes the following forms:—

II. *Multiply the weight by the radius in feet and by the square of the angular velocity; divide by gravity (32.2); or otherwise,*

III. *Multiply the weight by the radius in feet, and by the square of the number of revolutions per second; divide by 0.8154.*

If the number of revolutions *per minute* is used, the divisor becomes 2935.

81. *Centrifugal Force* is the re-action of a body moving in a curved path against the body which guides it, or compels it so to move; and according to the principles of Article 52 and Article 76, it is exactly equal and opposite to the Deviating Force exerted by the guiding body upon the body that is guided. Amongst the re-actions of the moving parts of an engine, which are to be balanced against each other, as stated in Article 76, the centrifugal forces of the several revolving parts are to be taken into account.

82. The instrument called a *Revolving Pendulum* is practically used in regulating the speed of machinery; but in this treatise the chief reason for referring to it is, that it is often convenient in calculations respecting the rolling movements of ships and of waves, to make use of the height of a revolving pendulum capable of keeping time with those movements.

In Fig. 42, CA represents a revolving pendulum, consisting of the bob, A, hung from the point, C, and swinging round the vertical axis, CB, in a circle whose radius is AB, perpendicular to CB. CB is called the *height* of the pendulum. The tension of the string or rod, CA, supplies at once the force which supports the weight of the bob, and the deviating force; so that we have the following proportion—

$$\text{Deviating Force} : \text{Weight} :: AB : CB.$$

But the deviating force is equal to—

$$\text{Weight} \times AB \times (\text{revols. per second})^2 \div .8154;$$

Therefore—

$$CB = \frac{\text{Weight} \times AB}{\text{Deviating force}} = \frac{.8154}{(\text{revols. per second})^2};$$

which result is expressed by the following rules:—

I. To find the height in feet of a revolving pendulum, which makes a given number of revolutions per second, *divide 0.8154 by the square of the number of revolutions per second; or otherwise, divide 2935 by the square of the number of revolutions per minute.*

II. To find the number of revolutions made in a given time by a revolving pendulum of a given height—

For revolutions per second; divide 0.8154 by the height in feet and extract the square root of the quotient:—

For revolutions per minute: divide 2935 by the height in feet, and extract the square root of the quotient.

83. *Oscillations* are movements performed by a body that is free to move, and yet stable, like a common swinging pendulum, or a ship afloat. When such a body is displaced from its position of balance, the righting force or couple (Art. 70) makes it swing back to that position, which it reaches with a velocity that carries it beyond, and makes it deviate to the other side of the position of balance; and that swinging to the one side and to the other would

Fig. 41.

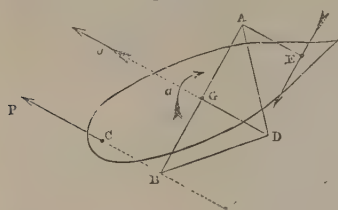
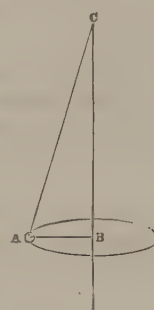


Fig. 42.



go on for ever, were it not gradually stopped by friction and other resistances.

By the *period of oscillation* of a body is to be understood, the time occupied in making a *complete oscillation* or *double swing*, at the end of which the body returns to the position that it started from; and when the *number of oscillations* per second, or per minute, is spoken of, complete oscillations are to be understood, except when *single oscillations* are specified.

When the righting force (or righting couple, as the case may be) is simply proportional to the disturbance (linear or angular, as the case may be) of the body from its steady position, the oscillations of the body, whether large or small, are performed in equal times, and are said to be *isochronous*. In such cases, the following rules serve to determine the height of a revolving pendulum (called the *equivalent pendulum*) which keeps time with the body, making the same number of revolutions in a given time that the body makes complete oscillations; from which height the number of oscillations in a second, or a minute, may be found by Rule II. of Art. 82:—

I. For linear displacement, or shifting—

As the righting force
: *is to the weight of the body*
: : *so is the extent of disturbance*
: *to the height of the equivalent pendulum.*

II. For angular displacement, or heeling—

As the moment of the righting couple (Art. 70)
: *is to the moment of inertia of the body's weight* (Art. 77)
: : *so is the disturbance in circular measure*
: *to the height of the equivalent pendulum.*

These principles will be illustrated in the sequel by their application to important questions connected with the rolling of

ships in smooth water, the rolling of the waves, and the movements of ships as affected by those of the waves.

When the righting force or couple is not simply proportional to the displacement from the steady position, large and small oscillations are not performed in equal times. It is unnecessary, however, to investigate this case in detail.

When a body oscillates in a resisting fluid, as a ship does in rolling amidst the water, the resistance causes the oscillations to become gradually less and less extensive until they stop. When the body in oscillating sweeps part of the fluid to and fro along with it, the oscillations are rendered slower than they would otherwise be; and such is the effect of a sharp floor and a deep keel on a ship's oscillations, as will be afterwards more fully shown.

84. *Re-action of Oscillating Bodies*.—An oscillating body re-acts on the bodies which exert the righting force or righting couple upon it, with an equal and opposite force or couple, which may accordingly be determined by reversing the proportional formulæ of Article 83; that is to say:—

I. For linear oscillations—

As the height of the equivalent pendulum
: *is to the greatest extent of disturbance*
: : *so is the weight of the body*
: *to its greatest re-action.*

II. For angular or rolling oscillations—

As the height of the equivalent pendulum
: *is to the greatest heel (in circular measure)*
: : *so is the moment of inertia of the body's weight*
: *to the moment of the greatest re-acting couple.*

These principles have an important bearing on the straining effect of the various oscillatory movements of ships.

CHAPTER III.

DISPLACEMENT AND STABILITY IN SMOOTH WATER.

85. *Subjects of this Chapter*.—The nature and use of the buoyancy of a ship, and of her stability in smooth water, having been explained generally in Chapter I., Sections 2, 3, and 4, it is the object of the present chapter to describe how the extent to which a given ship possesses those qualities is ascertained by detailed measurement and calculation, through the proper application of the principles of mensuration and mechanics set forth in Chapter II. The first section will relate to the determination of Displacement, and of the Centre of Buoyancy; the second, to the determination of stability by the approximate method, known as the *finding of the Metacentre*; the third, to the combination of calculations of displacement and approximate stability in one form, which will be illustrated by a detailed example; the fourth, to the more exact methods of computing stability which it is sometimes advisable to employ, also illustrated by detailed examples; and the fifth, to longitudinal stability in smooth water, and questions of trim depending on it.

SECTION I.—DISPLACEMENT, AND CENTRE OF BUOYANCY.

86. *General Description of the Plans of a Ship*.—The “*Draught*” or drawing of a proposed ship, from which measurements for finding the displacement and stability of the ship are made, consists of three plans of the external figure of the vessel, of which several examples are given in the Plates illustrating this treatise. The general nature of those plans is indicated on a small scale in the sketches, Fig. 1, Fig. 2, and Fig. 3.

The *SHEER PLAN*, Fig. 1, is a longitudinal elevation, or side view of the vessel, usually placed with the bow towards the right, and the stern towards the left.

The *HALF-BREADTH PLAN*, Fig. 2, is a horizontal plan of one-half of the vessel; which is sufficient, because the other half is exactly symmetrical to it.

The *BODY PLAN*, Fig. 3, consists of a pair of half-transverse elevations, or end-views of half the vessel, joined together in the

middle; so that the right-hand half represents the vessel as seen endwise from before, and the left-hand half as seen endwise from behind.

The sheer plan and half-breadth plan are always, and the body plan usually, drawn to the same scale, which ought to be large

enough for precise measurement. A quarter of an inch to a foot, or 1-48th of the real dimensions, is one of the scales commonly employed. For very large vessels it may be sometimes desirable to use a smaller scale, in order that the eye of the constructor may be able to take in the whole drawing at one view.

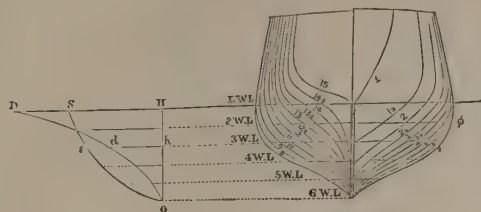


Fig. 4.

Fig. 3.

In the sketch at the head of this section the body plan, Fig. 3, is on a scale larger than the sheer and half-breadth plans, in the proportion of about 9 to 5, for the sake of distinctness.

A great variety of lines are shown on such plans. Some of those lines are specially connected with questions which will be considered further on in this Division; others are specially connected with the practical execution of the ship, and belong to the second and fourth Divisions of the treatise. In connection with displacement, two sorts of lines only have to be considered; and those are, WATER-LINES and VERTICAL CROSS SECTIONS.

A WATER-LINE is the outline of a horizontal section of the ship; being the line in which the surface of the water either actually meets the skin of the ship when she floats upright at a certain depth of immersion, or would meet the skin of the ship if she were to float immersed to a certain supposed depth.

In the sheer plan, Fig. 1, the water-lines are represented by horizontal straight lines. The uppermost, marked L.W.L., is the *load-water-line*, corresponding to the immersion of the ship when most heavily laden; the lowest (marked in the present case, 6 W.L., as being the sixth in order from the load-water-line inclusive) runs along the lower edge of the "*rabbit*," or groove, where the skin of the vessel joins the keel, if she is to float "on an even keel"; that is, with the keel horizontal. If the keel is to have an inclination, the lowest water-line makes a small angle with the keel. In vessels which have no keel, the lowest part of the ship's bottom is to be read instead of the lower edge of the rabbit of the keel.

The vertical depth between the highest and lowest water-lines is divided into a number of equal intervals by the intermediate water-lines, which are numbered in succession 2 W.L., 3 W.L., &c. If necessary in calculating the displacement, those intervals may be subdivided, according to the principles explained in Art. 19. This is often required at those parts of the vessel's bottom which are rapidly curved, such as the *bilge*, being the part which connects the nearly upright side with the comparatively flat *floor*. In Figs. 1, 2, and 3, there is no such subdivision; but an example of its use will be given further on.

On the Body Plan, Fig. 3, as well as on the Sheer Plan, the water-lines are marked by horizontal lines. It is on the Half-breadth Plan, Fig. 2, that their figures are shown; and this is the principal use of that plan.

The plane horizontal area inclosed within a water-line is called a *Water-section*, or a *Plane of Flotation*.

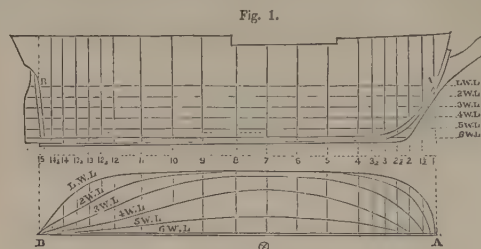


Fig. 1.

Fig. 2.

As a *base line*, or *longitudinal axis*, for all the measurements of the ship, there is taken the *Centre Line of the Load-water-section*, marked A B in the sheer plan and half-breadth plan, and represented by the point, A, in the body plan. It is held to extend from the forward edge of the rabbet of the stem at A, to the after edge of the rabbet of the stern-post at B, being the points where the surface of the vessel meets the stem and stern-post respectively; and that distance is divided into a sufficient number of equal intervals, which are subdivided where the figure of the vessel requires it.

When the vessel has two stern-posts, a main stern-post and a rudder-post, as is the case with most screw steamers, it is at the foremost of the two, or main stern-post, that the length of the load-water-section is held to terminate.

In the sketch, the base line is divided into fourteen intervals; and the three foremost and three aftermost intervals are subdivided into half intervals, because of the rapid curvature of the water-lines near the bow and stern. In vessels with fine entrances and runs to the water-lines, like most of the steam vessels whose plans are given in this treatise, such subdivision is unnecessary.

The *Cross Sections* are indicated in position on the sheer plan by vertical straight lines, and on the half-breadth plan by transverse lines, traversing the points of division of the base line.

The largest of those cross sections is called the *Midship Section*, or *Midship Bend*, and is marked with the symbol \otimes on the plans. It divides the ship into two parts, called respectively the *Fore Body* and the *After Body*. Sometimes there is a part of the ship in the middle which has an uniform cross-section throughout its length; its water-lines being parallel straight lines. This, when it exists, is called the *Middle Body*.

The Cross Sections, whose position only is shown on the sheer plan and half-breadth plan, have their figures shown on the body plan; the outline of which shows the midship section; its right-hand half, the half-cross sections of the fore body; and its left-hand half, the half-cross sections of the after body.

To enable the same cross sections to be found on the different plans, they are marked with numbers. The old method of numbering was, to mark the cross sections of the fore body with letters, and those of the after body with figures, beginning at the midship section; but according to the method now found to be most convenient, the cross sections are numbered in their order, from bow to stern—that at the commencement, A, of the longitudinal axis being No. 1, whole numbers being affixed to the cross sections at whole intervals, and fractional numbers to those at fractional intervals:

for example, in the sketches at the head of this Section, the cross sections are numbered 1, $1\frac{1}{2}$, 2, $2\frac{1}{2}$, 3, $3\frac{1}{2}$, 4, 5, 6, &c. The midship section happens to be No. 7 in the present case.

It is to be observed, that in consequence of No. 1 being allotted to the cross section at the starting-point, A, the *numbers of intervals* by which the cross sections are distant from that point, are each less by one than the numbers affixed to the sections. For example—

Nos. affixed to Cross Sections, 1, $1\frac{1}{2}$, 2, $2\frac{1}{2}$, &c.....	14, $14\frac{1}{2}$, 15 ;
Intervals from A, 0, $\frac{1}{2}$, 1, $1\frac{1}{2}$, &c.....	13, $13\frac{1}{2}$, 14 ;

and this must be attended to in computing moments, as will afterwards be illustrated.

In the same manner, from the load-water-line being considered as No. 1, and the others numbered 2, 3, 4, &c., it follows that the number of *vertical intervals* by which any water-section is distant from the load-water-section is less by one than the number affixed to the section; for example—

Water-sections; L.W.L.; 2 W.L.; 3 W.L.; &c.	
Intervals below L.W.L.; 0 ; 1 ; 2 ; &c.	

The parts of the ship's external figure under water, which are near the bow and stern respectively, are known as the *entrance* and the *run*. The intermediate part of the bottom, extending from the keel sideways, is called the *floor*, and is said to be *flat* or *rising*, according to its figure. The *bilges* are the curved parts of the bottom, which connect the floor with the sides.

That part of a ship's external figure which is sometimes under and sometimes above water, either from variations in the displacement, or from pitching and rolling, is said to be *between wind and water*, and its form is of primary importance as regards stability.

87. The *Ordinates of a ship* are all *half-breadths*; that is to say, horizontal lines, measured from the central vertical longitudinal plane traversing the axis AB (Fig. 1) to the outside surface of the vessel. The ordinates shown on the plans are situated at the intersections of the horizontal or water sections, and the vertical or cross sections; so that, in the example shown in the sketch, as there are twenty-one cross sections and six water-sections, there are $6 \times 21 = 126$ ordinates, or half-breadths.

The sheer plan shows the positions of all the ordinates, but not their lengths. The half-breadth plan shows all their lengths, and their positions lengthwise, but not vertically; the body plan shows all their lengths, and their positions vertically, but not lengthwise.

It is evident that any two of those three plans show the lengths and positions of all the ordinates completely, and are therefore, in a strict mathematical sense, sufficient to show the figure of the ship completely; but the three plans together give a more distinct idea of the ship's figure to the mind, and are more convenient for purposes of measurement and calculation.

Any two of the plans being given, the third can be constructed from them. The method of doing this will be explained in the Second Division.

Each ordinate belongs at once to the water-section and to the vertical section of which it is the intersection; and by Simpson's rules, it has two multipliers, according as it is to be used in computing the area of the water section, or that of the vertical section. For example, in the Figures, the ordinate at the intersection of 2 W.L., and vertical section, No. 6, belongs at once to each of those sections; and by Simpson's First Rule, its multiplier is 2 in computing the area of the second water-section, and 4 in computing

that of the sixth vertical section. If it were desired to compute the displacement in rectangular blocks, as described in Art. 26, the multiplier for each ordinate would be the product of its two multipliers for sectional areas: for example, in the case of the ordinate just mentioned the multiplier would be 8; but this method of calculation is seldom or never employed.

Occasionally some of the intersections of the water-lines and vertical sections on the sheer plan fall beyond the outline of the vessel at the bow or stern. Such intersections may be called *blank ordinates*. The true value of each of them is 0; but since in the parts of the ship where these occur the surface of the ship is nearly parallel to the vertical plane through the axis, AB, it is more nearly correct to set down in the calculations each blank ordinate as being equal to the real ordinate directly above it—e.g., the thickness of the stem, or stern-post, as the case may be—and afterwards to make a deduction for the excess so introduced into the displacement.

When a vessel floats much deeper at the stern than at the bow, so that the keel has a considerable slope, the series of water-sections may be continued, with suitable intervals, down to the lowest point of the rabbet of the keel, adjoining the heel of the stern-post; and then the lower water-sections will contain many blank ordinates, which will be treated in the way just mentioned.

88. *Methods of Computing Displacement.*—Setting aside the method by rectangular blocks as not being customary, there are two processes for computing the displacement of a ship, both of which should always be gone through; because the intermediate steps of both processes are necessary in the subsequent operation of finding the centre of buoyancy; and also because the agreement of their final results forms a check on the accuracy of the calculations.

One process consists in first computing, by Simpson's rules, the areas of the several vertical sections; and then treating those areas as the ordinates of a new curve upon the base, AB, in order to compute the volume of the displacement, by the method of Art. 24. Sometimes, though not always, that curve is drawn to a scale, and is called "*Peake's Curve*," or the *Curve of Sectional Areas*.

The other process consists in computing the areas of the several water-sections, and then treating those areas as the ordinates of a new curve upon a base equal to the distance between the load-water-line and the lowest water-line, in order to compute the displacement, not only up to the load-water-section, but up to each water-section of the series.

89. The *Curve of Water-sections* is represented in Fig. 4, by OsS; its base, OH, is the load draught of water, measured upwards, from the *lowest point of the keel*; its greatest ordinate, HS, represents the area of the load-water-section, and any other ordinate, such as hs , the area of the water-section at a less draught of water, such as Oh .

The area, OSH, represents the load displacement; and the area, Osh, the displacement at the draught of water, Oh .

The area of a given water-section represents also the *displacement in cubic feet per vertical foot of immersion* at that water-section; which, being divided by 35, gives the *displacement in tons per foot of immersion*. This again, being divided by 12, gives the *displacement in tons per inch of immersion*

$$= \frac{\text{Water-section in square feet}}{420}.$$

It is customary, in drawing the curve of water-sections on the plans of a ship, to lay down its horizontal ordinates, HS , hs , &c., to such a scale that they shall represent, not the areas of the water-sections themselves, in square feet, but the 420th parts of those areas, or the *tons per inch immersion*. The use of that quantity has already been illustrated in Chap. I., Art. 2. It enables us to calculate how much deeper a given ship will be immersed by a given addition to her lading.

90. *Curve and Scale of Displacements.*—The displacements themselves, in tons of 35 cubic feet, corresponding to different draughts of water, are laid down on the drawing as the horizontal ordinates of a curve, OdD . For example, the ordinate, HD , represents the load displacement, and the ordinate, hd , the displacement at the draught, Oh . A scale of tons is marked along the longest ordinate, HD .

91. *Computation of Cross Sections.*—As each vertical cross section consists of two similar halves, it is customary to begin by computing the half area of each vertical section, and afterwards to multiply by 2. The appearance of the vertical sections upon the body plan enables the naval architect to judge where and to what extent subdivision of the vertical intervals is required; and that subdivision should be made by means of intermediate water-sections running the whole length of the ship, for the sake of uniformity in the calculations, the neglect of which is apt to lead to confusion and mistakes.

91A. *Modifying Cross Sections.*—In making the first rough design for a ship, it may sometimes be desirable to alter the breadths and the figure of the side at and near the load-water-line,

without altering either the load displacement of the vessel, the distribution of that displacement longitudinally, or the total draught of water; for which purpose it is necessary that the area, up to the load-water-line, of each cross section, as well as the immersed depth of that section, should remain unchanged, although the figure of the section, and its breadth at the load-water-line, may be modified.

The following rule has been found convenient in such cases:—

In Fig. 4A, let $ABEHC$ represent one-half of a given cross section, AB being the load-water-line, and let it be required to increase the half breadth of that section from AB to AB' , without altering its area or its depth. Divide the immersed depth, AC , into three equal parts by the points D and F , at which draw the horizontal ordinates, DE and FH . Then the ordinate, DE , at one-third of the total immersed depth, is to be kept unchanged; and from the ordinate, FH , at two-thirds of that depth, is to be cut off HH' , equal to one-third of BB' , the quantity by which the ordinate at the load-water-line has been increased. Then let a new cross section be traced through the points, B' , E , H' , C ; and by Simpson's Second Rule (Art. 17, Rule III.) the area of the new section will be either exactly or approximately equal to that of the original section. When the two curves, $BEHC$ and $B'EHC$, are both parabolas, either of the second or of the third order, the equality will be exact.

92. *Computation of Water-sections.*—The water-sections, like the cross sections, consist of two similar halves; and therefore, in general, the half areas are computed first, and afterwards multiplied by 2. The process of measurement and calculation requires no

special remark, being almost always performed by means of Simpson's Second Rule, with subdivided intervals where they are required (Art. 19), of which the naval architect judges from the appearance of the half-breadth plan.

93. *Computation of Displacement in Layers.*—The computation of the load displacement presents no peculiarity; it is performed by treating the areas of the water-sections just as the ordinates are treated in computing areas of cross sections, the series of multipliers being exactly the same.

In computing the series of displacements up to the other water-sections, the particular rule employed must be varied according to the circumstances of the particular calculation. For example, to find the displacement up to 2 W.L., the water-line next below the load-water-line, the volume of the layer between those two water-sections is to be computed by the rule of Article 18A, and subtracted from the load displacement. (The rule referred to takes in this case the following form:—To five times the area of L.W.L. add eight times the area of 2 W.L., and subtract the area of 3 W.L.; multiply the remainder by one-twelfth of the vertical interval, or depth of the layer; the product will be the volume of the layer.) The volume of any even number of equally deep layers is to be computed by Simpson's First Rule, and that of three equally deep layers by Simpson's Second Rule (Art. 19).

Detailed examples of all those methods of calculation will be given in Section III. of this chapter.

94. The name of *Appendages* is given to small portions of the ship which project beyond the network of water-sections and cross sections, and whose volumes must therefore be found by special calculations, and added to the main part of the displacement. They usually consist of the keel below its rabbet, the false keel (if any), part of the stem, part of the stern-post, the rudder, and the rudder-post and screw in screw steamers. In vessels which float much deeper at the stern than at the bow, it may be convenient to take the lowest water-section higher than the lowest point of the rabbet of the keel, and then part of the "dead-wood," or lowest part of the run, will be treated as an appendage.

Amongst the appendages is also reckoned a *negative* or *subtractive appendage*; being the deduction to be made from the displacement in order to correct the effect of having assumed a certain value for the blank ordinates (see Art. 87) whose real value is 0. This deduction is usually called the "*part supposed*." The volume of the appendages is often so small (especially in iron ships), that it may be neglected without practical error.

95. *Computation of Midship Section in Layers.*—It is a common practice to compute the area of immersed midship section for a series of different draughts of water, like the displacement; and the process is perfectly analogous. The areas are then represented by the horizontal ordinates of a curve, which usually in general appearance is somewhat like the curve of displacement, OdD , Fig. 4. It was formerly supposed that the resistance of a given ship at the same speed, and at different immersions, varied proportionally to the area of the immersed midship section; but that supposition was founded on an imperfect theory of the resistance of fluids, and has not been corroborated by experience.

96. *Determination of Centre of Buoyancy.*—The nature of the centre of buoyancy, and the use of finding its position, have been explained in Art. 3 of Chap. I.

As the immersed part of a ship floating upright consists of two symmetrical halves, one on each side of the central vertical plane

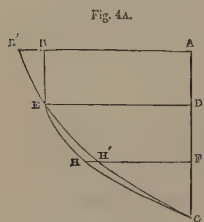


Fig. 4A.

which traverses A B, Figs. 1 and 2, it is obvious that the centre of buoyancy of a ship floating upright must be in that plane; so that in order to find the position of that centre completely, it is only necessary to find its horizontal distance from the plane of the cross section through A, and its vertical depth below the load-water-section.

To find the horizontal distance of the centre of buoyancy from a transverse vertical plane through A, the first step is to compute the *moment* of the volume of displacement relatively to that plane by the rule of Chap. II., Section II., Art. 38; that is to say, the area of each cross section is to be multiplied by its distance from A, and the products treated as the ordinates of a new curve. The moment thus found, being divided by the volume of the displacement, gives the distance required.

To find the depth of the centre of buoyancy below the load-water-section, the first step is to compute the moment of the volume of the displacement relatively to the plane of that section, by the rule just referred to; that is to say, the area of each water-section is to be multiplied by its depth below the load-water-section, and the products treated as the ordinates of a new curve. The moment thus found, being divided by the volume of the displacement, gives the depth required.

In performing these calculations, time is saved by the method already explained in Art. 38, of multiplying the sectional areas in the first instance, not by the leverages themselves, but by the numbers of intervals to which those leverages are proportional, and performing a multiplication by the common interval after the addition has been made. Time and trouble may also be saved by observing, that in computing the displacement, the sum of certain products is multiplied by one-third of the common interval; and that in computing the moment, the sum of certain other products is also multiplied by one-third of the common interval; so that when the object is simply to obtain the quotient of the division of the moment by the displacement, no error will arise from omitting the multiplication by one-third of the common interval in computing both dividend and divisor. This will be illustrated by examples in Section III.

96 A. *Computations from plans with Cross Sections oblique to the Water-sections.*—It may be necessary in certain cases to compute the displacement and find the centre of buoyancy of a ship, from a set of plans in which the cross sections are not exactly vertical, but are perpendicular to a keel which makes a certain angle with the horizon. The construction of such plans, and the relation which they bear to plans with the cross sections vertical, will be explained in the Second Division of this treatise. In such cases the following rules are to be observed:—

I. In computing the *areas* both of cross sections and water-sections, the intervals between the ordinates are to be measured as already described in the preceding articles; that is, for cross sections, down the centre line of each section, or perpendicular to the keel; and for water-sections, parallel to the centre line of the load-water-section.

II. In computing the displacement by means of the areas of cross sections, the intervals between the cross sections are to be measured in a direction perpendicular to those sections; that is, *parallel to the keel*. To reduce intervals measured horizontally to the corresponding intervals measured parallel to the keel, *Multiply by the cosine of the angle of inclination of the keel*.

III. In computing the displacement by means of the areas of water-sections, or in horizontal layers, the intervals between

the water-sections are to be measured perpendicular to those sections; that is, *vertically*. To reduce intervals measured perpendicularly to the keel to the corresponding vertical intervals, *multiply* (as before) *by the cosine of the angle of inclination of the keel*.

IV. Time may be saved by postponing the multiplication by the cosine just mentioned, until the end of the whole calculation; that is to say, *compute the displacement as if the cross sections were exactly vertical, and multiply it by the cosine of the angle of inclination of the keel*. The same rule applies to all the layers into which the displacement may be divided.

V. To compute the cosine of the angle of inclination of the keel in the absence of tables and of angular measurements:—*From the square of the length on the load-water-line subtract the square of the difference of depths of immersion of the two ends of a line along the rabbet of the keel; the square root of the remainder will be the total length measured parallel to the keel. Divide this by the length on the load-water-line; the quotient will be the cosine required.*

VI. In calculating the position of the centre of buoyancy, the simplest method is to omit altogether the multiplication by the cosine, both from the calculations of moments and from that of displacement; observing, that in laying down on the plan the co-ordinates thus computed, the depth below the load-water-section is to be drawn, not vertically, but in a direction *perpendicular to the keel*.

VII. In consequence of the obliquity of the line so drawn, the position of the centre of buoyancy is shifted horizontally from where it would otherwise have been, to a distance which is found by *multiplying the oblique depth below the load-water-section by the sine of the angle of inclination of the keel*; which sine is found by dividing the difference of depths of immersion mentioned in Rule VI. by the length on the load-water-line.

97. *Coefficients of Fineness.*—If two ships have figures so far similar, that every ordinate or half breadth in one of them bears an uniform proportion to the corresponding or similarly situated ordinate in the other ship, it is evident that the displacements of those two ships will be to each other simply in the proportion of the products of their lengths, extreme breadths, and immersed depths or draughts of water; that is, of the rectangular solids circumscribed about their respective immersed bodies.

Hence, if it has been ascertained that the displacement of a given ship is a certain fraction of the circumscribed rectangular solid, the displacement of any other ship of similar figure (as above defined) may be found by multiplying the product of her length, extreme breadth, and immersed depth, by the same fraction. That fraction is called a *coefficient of fineness*; because, by being greater in ships with bluff ends and flat floors, and smaller in ships with fine ends and rising floors, it furnishes a sort of indication of the general character of a ship's figure.

Examples of the coefficient of fineness will be given in the sequel. Amongst its commonest values are those which range from 0.5 to 0.66; but it is occasionally as low as 0.3, and as high as 0.8.

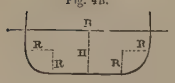
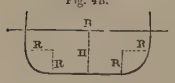
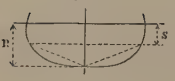
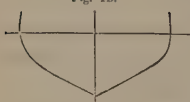

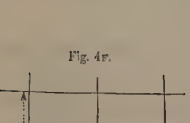
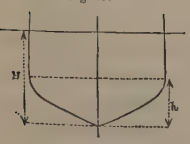
Besides the just-mentioned coefficient of fineness of the displacement, coefficients of fineness may also be computed for cross sections and for water-sections. Thus the midship section, being divided by the rectangle of its extreme breadth and immersed depth, gives a coefficient which ranges from 0.5 to very near 1.

The coefficients of fineness of water-lines, obtained by dividing the area of a water-section by the rectangle of its length and extreme breadth, range in extreme cases from 0.5 to 0.9, the more common proportions being from 0.6 to 0.75.

The mean coefficient of fineness of all the water-lines of a ship is obtained as follows: multiply the greatest immersed area of midship section by the length of the load-water-line, and divide the load displacement by the product.

The coefficient of fineness of the displacement is equal to the product of the coefficient of fineness of the midship section, multiplied by the mean coefficient of fineness of the water-lines.

TABLES OF COEFFICIENTS OF FINENESS OF CERTAIN SPECIAL FIGURES.

CROSS SECTIONS.	COEFFICIENTS.
Rectangle, 	1
Rectangle rounded with quadrants at the bilges; breadth, B; draught of water, H; radius of each bilge, R; (Fig. 4b), 	$1 - 4.292 \frac{R^2}{B H}$
Semicircle or semi-ellipse, immersed up to a diameter (Fig. 4c), 	.7854
Semicircle or semi-ellipse, with a segment cut off at the bottom, so as to make a flat floor; radius of semicircle, or vertical semiaxis of semi-ellipse, R; draught of water, S (the arc to be taken in circular measure—see Art. 30, p. 15)— $\frac{R}{2S} \text{ arc. sin. } \frac{S}{R} + \frac{1}{2} \sqrt{1 - \frac{S^2}{R^2}}$	
Semicircle or semi-ellipse, with a pair of segments cut off at the bottom, so as to make a sharp floor; radius of semicircle, or vertical semiaxis of ellipse, R; depth from the water-line to the upper corners of the segments, S— $\frac{1}{2} \text{ arc. sin. } \frac{S}{R} + \frac{1}{2} \sqrt{1 - \frac{S^2}{R^2}}$	
Triangle, 	$\frac{1}{3}$
Pair of Common Parabolas, vertices at the water line (Fig. 4e), 	$\frac{2}{3}$
Pair of Cubic Parabolas, vertices at the water-line (Fig. 4f), 	$\frac{2}{5}$
Pair of Parabolas of the nth order, vertices at the water-line, $\frac{n}{n+1}$	
Compound Straight and Parabolic—viz., vertical straight lines near the water-line; bilges and floors, a pair of common parabolas; total draught of water, H; depth of parabolic part, h (Fig. 4g), 	$1 - \frac{h}{3H}$
WATER-LINES.	COEFFICIENTS.
Rectangle, Ellipse, Parabolas of various orders, with their vertices at the extreme breadth:* see the preceding Table.	
Two Pairs of Parabolas of the third order, with their points of contrary flexure at the bow and stern,	$\frac{2}{3}$

* A proposed method of using parabolas of various orders for the lines of ships is described by Mr. Nyström, in a series of papers published in the *Arkiv* for May, 1863, and subsequently.

WATER-LINES.	COEFFICIENTS.
Pair of Harmonic Curves, complete (see Art. 22, Fig. 13),...	$\frac{1}{2}$
Pair of Harmonic Curves, convex part only,637
Pair of Trochoids (see Art. 22, Fig. 14); Length, L; Midship breadth, B,	$\frac{1}{2} + .3927 \frac{B}{L}$
Fore body a pair of harmonic curves; length of fore body, L_1 ; Middle body, a pair of parallel straight lines; length of middle body, $L - \frac{2}{3} L_1$; After body, a pair of trochoids; length of after body, $\frac{2}{3} L_1$ (being the form introduced by Mr. Scott Russell); Midship breadth, B; Total length, L ,	$1 - \frac{2}{3} \frac{L_1}{L} + .1963 \frac{B}{L}$
Lissoneid (described in a paper read to the Royal Society in 1863),634

The construction of the curves mentioned in the preceding Tables will be explained in the Second Division of this treatise.

98. The *Mean Depth of Immersion* of a ship's hull is computed, according to the principles of Chap. II., Section I., Art. 27, by dividing the displacement by the area of water-section. Its use will be explained in a later chapter.

99. The *Tonnage* of a ship, according as the word is qualified, may mean either the Displacement in Tons, the Burden, the Registered Tonnage, or the Tonnage by "Builders' old measurement:—"

I. The *Displacement* has been sufficiently explained.

II. The *Burden* means, the number of tons of lading which the ship is able to carry, in addition to the weight of her hull and equipments. It is obviously equal to the difference between the displacement when light, and the displacement when loaded; and on the scale of displacement (Fig. 4), supposing OH to represent the load draught of water, and Oh the light draught, the burden is represented by the difference between the ordinates, HD and $h d$. Hence the burden of a ship whose dimensions and figure, and light and load draughts are given, can always be calculated with precision.

According to what has already been stated in Chap. I., Art. 2, the burden of a ship ranges from about one-half to two-thirds of her load displacement, according to the heaviness or lightness of her construction. Wooden ships are heavier than iron ships of the same load displacement, and ships of war heavier than merchant ships. The following may be taken as ordinary proportions:—

	Per Cent. of Load Displacement.	
	Ship.	Lading.
Iron Merchant Ships, ...	35	65
Wooden Merchant Ships,	40	60
Ships of War, from	40	60
" " to	50	50

For reasons which will appear in the Third Division of this treatise, large ships, to be equally strong with small ships, must be made proportionally heavier; so that the weight of a large ship will form a greater percentage of the displacement, and that of a small ship a smaller, than the average stated above. This applies especially to the skin, the keel, and all longitudinal parts of the framing, whose weight should vary nearly as the displacement multiplied by the length, or, in similarly shaped vessels, as the displacement multiplied by its own cube root.

The total burden of a steam-vessel includes her engines and store of fuel; hence, to find her net burden, available for cargo, those weights must be subtracted from the total burden. The proportion which they bear to the displacement varies very much in different cases, according to the speed, the figure of the ship, and the construction and efficiency of the engine; and it is likely to undergo very great diminution when improvements in design

and in economy shall have been generally adopted in practice. The results of present practice, with moderately good design and economy, may be roughly approximated to by the following rules:—

A steamer of 1000 tons' displacement, to go at a full speed of ten knots under steam, requires engines of the weight (including boilers) of about 125 tons, and a store of coal of one ton per hour of the voyage, or one-tenth of a ton per nautical mile.

The weight of engines, and of fuel consumed *per hour*, varies nearly as the square of the cube root of the displacement, and as the cube of the speed; but the weight of fuel *for a given trip* varies as the square of the speed.

[In algebraical symbols, let V denote the speed in knots per hour, and D the displacement in tons; then—

$$\text{Tons' weight of engines and boilers} = \frac{V^3 D^3}{800} \text{ nearly;}$$

$$\text{Tons coal per hour} = \frac{V^3 D^3}{100,000} \text{ nearly;}$$

$$\text{Tons coal per nautical mile} = \frac{V^3 D^3}{100,000} \text{ nearly.}]$$

Such calculations as these, however, give but a loose approximation; for the actual weight of engines and boilers, even according to ordinary examples, may range from four-fifths to once and a quarter of that given by the above rule, and the consumption of fuel within even wider limits (say from two-thirds to once and a half in ordinary cases, and from half to double in extreme cases), owing to the great differences in the economy of engines. These matters will be further considered in treating specially of steam-power for navigation.

The burden of a ship may be computed approximately, by multiplying the area in square feet of the water-section midway between the load and light water-lines, by the difference between the load and light draught in feet, and dividing by 35 for tons. The area of that water-section may also be approximated to with considerable accuracy by a practised measurer, by measuring simply its extreme length and breadth, and multiplying their product by a coefficient of fineness, estimated by the eye (Art. 97); and this was the method of measuring the burden of ships, introduced by Chapman into Sweden.

III. *Registered Tonnage* means, an approximate estimate of the carrying power of the ship, made by means of a mode of measurement prescribed by law, for purposes of taxation. The present mode of ascertaining the registered tonnage of ships, established in Britain by the Merchant Shipping Act of 1854, and by Regulations subsequently issued by the Board of Customs under the powers of that Act, consists in measuring the internal capacity of the vessel in cubic feet by Simpson's First Rule, dividing by 100 for tons, and deducting the space occupied by the propelling power, if any. The following is a summary of the rules to be observed (reference must be made to the Act and Regulations themselves for the details):—

The *tonnage deck* to be the upper deck in ships with fewer than three decks, and the second deck from below in all others.

Measurements to be expressed in feet and decimals.

The length is measured upon the tonnage deck from inside to inside of the inner plank at the stem and stern respectively. Deductions are made for the rake of the stem and stern in the thickness of the deck.

This length to be divided into equal intervals, according to the following rules:—

LENGTH	INTERVALS.
Not exceeding 50 feet,.....	4
Exceeding 50 feet, and not exceeding 120,.....	6
“ 120 “ “ 180,.....	8
“ 180 “ “ 225,.....	10
Exceeding 225,	12

Transverse sections are then measured at each point of division of the length, as follows:—

The total depths of the transverse sections are measured from a depth below the underside of the tonnage deck, equal to one-third of the round of the beam, to the upper side of the floor timbers, making a deduction for the average thickness of the ceiling (or internal skin of the hold).

Each such total depth is divided into intervals, according to the following rule:—

MIDSHIP DEPTH	INTERVALS.
Not exceeding 16 feet,.....	4
Exceeding 16 feet,.....	6

Breadths are measured horizontally at the points of division of the depths, and also at the highest and lowest points of each depth, “extending each measurement to the average thickness of that part of the ceiling which is between the points of measurement.”

The transverse areas are then computed by Simpson's First Rule (Art. 19); and thence the capacity of the ship is completed by the same rule (Art. 24); and the capacity in cubic feet being divided by 100, gives the registered tonnage. Fractions of a ton are expressed in hundredths; that is, in cubic feet.

The capacity of the poop, deck-houses, and other permanently inclosed spaces on the upper deck, available for cargo, passengers, or crew, is to be measured and included in the computation of tonnage, with the following exemptions—space for berthing the crew, not exceeding one-twentieth of the remaining tonnage; buildings for the shelter of deck passengers.

The space occupied by the propelling power, and by passages for the admission of light and air to the engine-room and boilers, is to be measured and deducted from the tonnage.

The term *gross tonnage*, in the case of steam-vessels, is applied to the tonnage as computed from the whole internal capacity, before deducting the space occupied by the engines. It appears that on an average of several iron screw merchant steamers, the gross tonnage is 0.62 of the displacement to 7-10ths of the moulded depth, measured from the gunwale to the rabbet of the keel.

IV. The *Tonnage by Builders' Old Measurement* (abbreviated b. o. m.) means, the tonnage measured by the rule which was formerly established by law in Britain, until repealed in 1836. It is a purely arbitrary function of certain dimensions of the ship, bearing no definite relation to any of her real qualities. Its use, however, is still retained by builders and purchasers of ships in their transactions. The following is the rule for computing it:—

Measure the length of the ship along the rabbet of the keel, from the back of the main stern-post to the foot of a perpendicular let fall from the fore part of the main stem. Measure, also, the extreme breadth, from outside to outside of the planking, exclusive of doubling planks. From the length as measured subtract *three-fifths* of the extreme breadth; the remainder is called the *keel for*

tonnage. Multiply the keel for tonnage by half the square of the extreme breadth, and divide by 94; the quotient will be the tonnage, b. o. m.

Fractions of a ton, b. o. m., are expressed in 94th parts.

The origin of this rule is somewhat obscure; but it was probably based on the following assumptions:—

That the load draught is one half of the extreme breadth;

That the coefficient of fineness for the displacement is 0.62;

That the burden is $\frac{2}{3}$ of the load displacement; from which it would follow that—

$$\text{Tonnage} = \frac{LB^2}{2} \times .62 \times \frac{2}{3} \div 35 = \frac{LB^2}{2 \times 94}.$$

The deduction of $\frac{2}{3}$ of the breadth from the length is believed to have been intended to compensate for the error introduced by including the rake of the stem in the measurement of length. The including of the rake of the stem and not that of the stern-post, it seems almost impossible to account for; unless it be that raking stern-posts were first invented after the enactment of the old tonnage law, as a device for evading its operation.

If the length for tonnage, b. o. m., were measured on the water-line instead of the keel, and the deduction of three-fifths of the breadth done away with, it would form a sort of rude approximation to the comparative power of different vessels to carry sail (as will afterwards appear); and hence there may be some reason for the practice of using it in regulating the allowances to be made for difference of size in races of sailing yachts.

The evil effects of the old tonnage law on the efficiency and safety of ships are well known. Since its repeal, they have almost passed away; but a trace of them is still to be found in the shape of a lingering prejudice on the part of some purchasers of ships, that it is an advantage for a ship to have a large actual burden compared with her tonnage by old measurement; or in other words, a small tonnage by old measurement for her actual burden. The fact is exactly the reverse; for as small tonnage b. o. m. implies small power of carrying sail, such a ship would be unstable, and a bad sailer—the very faults which the old tonnage law tended to encourage.

SECTION II.—APPROXIMATE CALCULATION OF STABILITY.

100. *General Explanations.*—The mode of operation of the forces which give stability to a ship, has been stated generally in Chapter I., Article 4; and the general subject of stability has been further explained in Chap. II., Article 70. In the present Section will be explained the method of calculating approximately the resistance of a ship to heeling, which is employed in ordinary cases, and known as the “metacentric method.” This will be illustrated by numerical examples in Section III. of the present chapter; and in Section IV. will be explained those more exact and detailed methods of computing stability which are sometimes required.

The stability of a vessel is expressed as a quantity by means of the *proportion which the moment of the righting couple bears to the sine of the angle of heel* (Chap. I., Art. 4). In every ordinary case in practice, that moment consists of the *difference* of two quantities, a righting couple and a smaller upsetting couple, which are produced as follows:—

Let Fig. 5 represent the end view of a ship, G her centre of gravity, GZ her *upright axis*, Y'Y her water-line, and C her centre of buoyancy, when in an upright position. Instead of making a new drawing of the ship in an inclined position (as has been done

in the figures which illustrate Article 4), it is more convenient to suppose the same drawing to be heeled over, and to lay down upon it an *inclined water-line* W'W; the force of gravity being now supposed to act in the direction GZ', perpendicular to the inclined water-section.

The upright and inclined water-sections cut each other in a longitudinal straight line (whose projection on the plane of the drawing is represented by S); and the position of that line is determined by the fact, that the displacement after heeling is the same as before.

Those two water-sections contain between them a pair of wedge-formed solids, YSW, and Y'SW'. The solid, YSW, is called the *wedge of immersion*; because, in the act of heeling, it is plunged into the water, and added to the displacement. The solid, Y'SW', is called the *wedge of emersion*; because, in the act of heeling, it is lifted out of the water, and taken from the displacement. The volume of the displacement remains unchanged; therefore the volumes of the wedges of immersion and emersion are equal.

(Those wedges, for brevity's sake, are sometimes called simply the “*in*” and the “*out*.”)

The change of figure which the volume of water displaced undergoes in the act of heeling, may be regarded as being produced by *transferring* the solid, Y'SW', to the position, YSW. Hence the centre of buoyancy is shifted into a new position, which may be found according to the principles of Chapter II., Article 41. That is to say, let I be the centre of the wedge of immersion; E, that of the wedge of emersion; through the original centre of buoyancy, C, draw CC' parallel to EI, and make the distance, CC', less than the distance, EI, in the same proportion in which each of the wedges is less than the whole displacement; that is to say, make

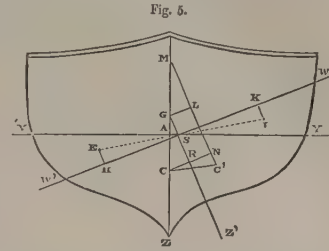
$$\overline{CC'} = \frac{\overline{EI} \times \text{volume of wedge}}{\text{Displacement}};$$

then C' will be the new position of the centre of buoyancy.

The line of action of the resultant pressure of the water, in the inclined position of the ship, is C'M, perpendicular to the inclined water-line, W'W; that is, parallel to GZ', and cutting GZ in M (which last point is, under certain circumstances to be afterwards specified, called the *Metacentre*); and the *righting couple* consists of that resultant pressure, acting upwards, and of the weight of the ship acting downwards through her centre of gravity, G, in the direction, GZ'; so that the arm of that couple is the perpendicular distance, GL, of the centre of gravity, G, from C'M.

Through the original centre of buoyancy, C, draw CN perpendicular to C'M; that is, parallel to W'W, and cutting GZ' in R. Then the moment of the righting couple may be regarded as the *difference* of the two following moments:—

The *Moment of Surface Stability*, Displacement $\times \overline{CN}$, being the moment due to the alteration of the position of the centre of buoyancy, and being what the righting moment would be if the centre of gravity, G, coincided with the original centre of buoyancy, C; from which has to be deducted—



Displacement $\times \overline{CR}$; being the diminution of the righting moment caused by the fact, that the centre of gravity is above the original centre of buoyancy.

Observing that the displacement enters as a common factor into both those moments, we have, for the righting couple, the following formula:—

(I.) Displacement $\times (\overline{CN} - \overline{CR})$; and this will be expressed in cubic feet of water at a leverage of one foot, or in foot-tons, according to the unit in which the displacement is stated.

Another formula for the same moment is arrived at as follows—

From E and I, the centres of the wedges of emersion and immersion, let fall perpendiculars EH and IK on the inclined water-section, WW'. Then the moment of surface stability (Displacement $\times \overline{CN}$), may be also expressed thus:—

Transferred Wedge $\times \overline{HK}$;

and this quantity is obviously equal to the *arithmetical sum of the moments of the wedges of immersion and emersion relatively to the axis, S, in which the upright and inclined water-sections cut each other*. Hence the moment of the righting couple may be expressed as follows:—

(II.) Moment of Wedges — (Displacement $\times \overline{CR}$).

Agreeably to what has been stated at the commencement of this Section, the measure of the stability of a given vessel as compared with others is to be found by dividing the righting moment at a given angle of heel by the sine of that angle. Now in Fig. 5, the angle of heel is ZGZ' = CM C' = WSY, and its sine is equal to each of the following ratios—

$$\frac{\overline{CN}}{\overline{CM}} = \frac{\overline{CR}}{\overline{CG}} = \frac{\overline{GL}}{\overline{GM}}.$$

When each of the expressions (I.) and (II.) for the righting moment is divided by the sine of the angle of heel, the result is as follows, for the measure or coefficient of stability:—

(III.) Displacement $\times \overline{GM}$
= Displacement $\times (\overline{CM} - \overline{CG})$;

the height, CM, of the point, M, above the original centre of buoyancy, being found by the following formula:—

(IV.) $\overline{CM} = \frac{\text{Moment of Wedges}}{\text{Displacement} \times \text{sine of angle of heel}}$.

101. *Metacentre*.—It is evident that the ship is *stable* or *unstable*, according as her centre of gravity, G, is *below* or *above* the point, M, where the upright axis of the ship, GZ, is cut by a vertical line traversing C', the disturbed centre of buoyancy; and that in ships of equal displacement, the stability is proportional to the height of M above G. The position of the point, M, therefore, regulates the distribution as regards level, of the weights of the ship and her lading, which is necessary in order to insure sufficient stability.

The already-mentioned name of *Metacentre*, in its strict acceptance, is given to the point, M, in two cases only—when it is a fixed point in the vessel for different angles of heel; and when it is determined by supposing the angle of heel indefinitely small. When the position of the point, M, varies for different angles of heel, its proper designation is the *Shifting Metacentre*.

There are few vessels in which M is an absolutely fixed point; but in well-formed sea-going vessels (for reasons which will be explained in Chapter IV.) its position varies very little at different angles of heel; and in them, therefore, it may be assumed to be a fixed point, without material error; and the present Section relates to the methods employed for ascertaining its position according to

that assumption, by computing its height above the centre of buoyancy, C, which is supposed to have been found by the methods of Art. 96 of the preceding Section.

It will also be explained how, when the metacentre has been found by calculation, the centre of gravity, G, may be found by experiment.

It is believed that in most sea-going vessels, the centre of gravity is nearly at the level of the load-water-line. The height of the metacentre above the load-water-line usually ranges from *three to six feet*, being greatest in the smallest vessels; and its average height is about *four feet*. When the metacentre is too low, the vessel is “crank;” when too high, she is “uneasy,” for reasons which have been briefly referred to in Chap. I., Articles 5 and 6, and which will be further explained in the fourth Chapter of this Division.

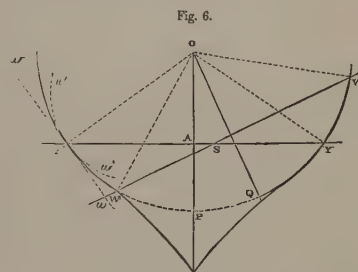
Assuming the metacentre to be fixed, the righting moment for any angle of heel is found by multiplying together the *displacement, the height, \overline{GM} , of the metacentre above the centre of gravity, and the sine of the given angle*.

Upon this principle, coupled with the fact that a common value for the height, \overline{GM} , is about four feet, is founded the statement already made in Chapter I., Art. 4, that “a common value for the moment of stability in large vessels, at an angle of heel of fifteen degrees, is the weight of the ship acting with a leverage of one foot.”

102. *Metacentre of a Vessel with Circular Sections*.—The only case in which the metacentre is an *exactly* fixed point in the vessel for different angles of heel, is that in which all the cross sections of the vessel “between wind and water” are arcs of circles described about one longitudinal axis. Vessels of this form are not usual in practice; but the method of finding their metacentres exactly is applicable to the finding of the metacentres of vessels of other forms approximately.

Let Fig. 6 represent a cross section of such a vessel, in which cross section the parts between wind and water are arcs of a circle described about the

point, O; and let it be required to find the moment of surface stability of a portion of such a vessel, measuring one foot in length. Let Y'Y be the upright, and W'W the inclined water-section. In the act



of heeling, the wedge of emersion, Y'SW', is taken from, and the equal wedge of immersion, YSW, added to the displacement; and if we consider that part of the displaced volume which lies within the cylindrical surface, Y'W'PQYW, this operation is equivalent to that of substituting the segment, W'QW, for the equal segment, Y'PY; or what is the same thing, that of turning the said segment about a longitudinal axis traversing O from the position, Y'PY, into the position, W'QW. Hence the required moment of surface stability is equal to the difference between the moments of the equal segments, Y'PY and W'QW, relatively to the longitudinal plane passing through OQ, perpendicular to the inclined water-section. But the moment of the segment, W'QW,

relatively to that plane, is nothing; and the moment of the segment, $Y'PY$ (according to Chap. II., Section II., Article 42) is $\frac{2}{3} \bar{A} \bar{Y}^2 \times \sin. POQ$;

which result is expressed by the following rule:—

I. *The moment of surface stability, per foot of length, is equal to two-thirds of the cube of the half breadth, multiplied by the sine of the angle of heel.*

The surface stability of the whole vessel is the sum of the surface stabilities of all her vertical layers, computed according to the Rule of Article 40, for finding the moments of wedge-formed solids, whence are deduced the following Rules:—

II. *Divide the length of the water-section into a convenient number of intervals, and measure the half breadths; treat the cubes of those half breadths as if they were the ordinates of a new curve (by the rules of Chap. II., Section I., Article 19); two-thirds of the area of that curve, multiplied by the sine of the angle of heel, will give the surface stability; and also—*

III. *Two-thirds of the same area, divided by the volume of displacement, will give the height of the Metacentre above the Centre of Buoyancy (CM, Fig. 5).*

[The algebraical expressions of these rules are as follows:—Let an indefinitely small interval of length be denoted by dx ; the corresponding half breadth or ordinate of the water-section, by y ; the angle of heel, $POQ = YSW$, by θ ; then—

$$\text{Moment of surface stability} = \frac{2 \sin. \theta}{3} \int y^3 dx; \text{ and}$$

$$\text{Height of Metacentre, CM} = \frac{\frac{2}{3} \int y^3 dx}{\text{Displacement}}.]$$

The moments computed by Rules I. and II. are expressed in cubic feet of sea-water at a leverage of a foot; they may be reduced to foot-tons by dividing them by 35.

From the explanation given in Chap. II., Section II., Article 43, it appears that two-thirds of the area whose ordinates are the cubes of the half breadths of the water-section, is the quantity called the *geometrical moment of inertia* of that section; hence Rule III. may be thus expressed:—*Divide the moment of inertia of the water-section by the volume of the displacement; the quotient will be the height of the metacentre above the centre of buoyancy.*

103. *Metacentre for Indefinitely Small Angles of Heel.*—It will now be shown that the position of the metacentre, which is exact for finite angles of heel in vessels of a certain figure, is also exact for vessels of any figure when the angle of heel is indefinitely small; in other words, that it is the *limiting position* to which the point, M, comes continually closer and closer, as the angle of heel is indefinitely diminished.

When that angle diminishes indefinitely, the following approximations take place at the same time:—

The line, S (Fig. 5), in which the upright and inclined water-sections intersect, approaches continually closer and closer to the longitudinal axis, A, of the upright water-section.

The cross sections, $Y'SW$, YSW , of the wedges of emersion and immersion, approximate closer and closer to equality with a triangle, which has its base equal to the half breadth, AY , its height equal to that half breadth multiplied by the sine of the angle of heel, and its area equal to the same sine multiplied by half the square of the half breadth.

The arithmetical sum of the moments of the cross sections of those wedges approximates closer and closer to double the moment of the said triangle relatively to the vertical plane through GZ ;

which double moment is equal to the area of the triangle multiplied by once and a third the half breadth—that is, to *two-thirds of the cube of the half breadth, multiplied by the sine of the angle of heel.*

Hence, according to the principles of Article 40, the moment of surface stability at an indefinitely small angle of heel is to be found by taking the cube of each of the series of half breadths of the upright water-section, treating those cubes as the ordinates of a curve, and multiplying two-thirds of the area of that curve (in other words, the moment of inertia of the water-section) by the sine of the angle of heel, being precisely the result arrived at in the preceding article; and hence the position of the metacentre found by the rule of that article is the limiting position to which the point, M, in a vessel of any shape approximates when the angle of heel is indefinitely diminished.

[The algebraical expression of this is as follows:—

Let $d\theta$ denote an indefinitely small angle of heel, in circular measure (Art. 40); the same symbol will denote the sine of that angle; because when an arc diminishes indefinitely, it approaches continually closer and closer to equality with its own sine.

Let dx denote an indefinitely small interval of the vessel's length; y , the corresponding half breadth of the water-section.

Then the cross section of each of the wedges of emersion and immersion is a triangle whose base is y , its height $y d\theta$, its area $\frac{1}{2} y^2 d\theta$, and its moment relatively to GZ , $\frac{1}{2} y^2 d\theta \times \frac{2}{3} y = \frac{1}{3} y^3 d\theta$; and the arithmetical sum of the moments of the wedges of emersion and immersion for the interval dx is—

$$\frac{2}{3} y^3 dx d\theta;$$

so that the moment of surface stability for the indefinitely small angle of heel, $d\theta$, is—

$$d\theta \times \frac{2}{3} \int y^3 dx;$$

and the height of the metacentre above the centre of buoyancy is—

$$CM = \frac{d\theta \times \frac{2}{3} \int y^3 dx}{\text{Displacement} \times d\theta} = \frac{\frac{2}{3} \int y^3 dx}{\text{Displacement}}.$$

as already found in Article 102.]

104. *Approximate Metacentre.*—From what has been stated in the preceding article, it appears that the position of the metacentre founded on the supposition that the vessel's cross sections between wind and water are of the kind shown in Fig. 6, and that she therefore has a fixed metacentre, is always an approximation to the shifting metacentre of a vessel of any form, being the limit of the positions of that point. The direction in which that approximation errs in any given case is determined by the following considerations:—

I. If the cross sections between wind and water have on the whole a *sharper convexity* than those giving a fixed metacentre (as shown by the dotted line, $w'w'$, in Fig. 6), then for any finite angle of heel, the actual wedges of immersion and emersion will be smaller than the assumed wedges; the actual moment of surface stability smaller than the assumed moment; and the actual position of the shifting metacentre will be *below* the approximate position; the error, in short, will be on the unsafe side. This case can scarcely ever occur in a skilfully designed vessel.

II. If the cross sections between wind and water are *concave*, or *straight*, or have on the whole a *flatter convexity* than those giving a fixed metacentre (as shown by the dotted line, ww , in Fig. 6), then, for any finite angle of heel, the actual wedges of immersion and emersion will be larger than the assumed wedges; the actual moment of surface stability will be greater than the assumed

moment; and the actual position of the shifting metacentre will be *above* the approximate position; in short, the error will be on the safe side. Such is almost always the case in skilfully designed vessels; and hence the practical utility of the metacentric method of determining the stability of a ship.

105. *The Method of finding a Ship's Centre of Gravity by Experiment* is founded on the principle set forth in Chapter II., Section II., Article 41, and Chapter II., Section III., Article 62, that if the centre of gravity of any part of a body be shifted in a given direction through a given distance, the centre of gravity of the whole body is shifted in a parallel direction through a distance smaller than the given distance, in the same proportion that the weight of the shifted part is smaller than the weight of the whole body.

In order to apply this principle to a ship, it is necessary that her metacentre should have been found by calculation. Then let

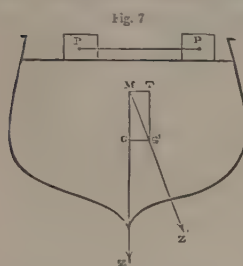


Fig. 7 represent a cross section of the ship, M her metacentre, and MZ her upright axis. It is already known that her centre of gravity is somewhere in that axis; but its depth below the metacentre remains to be ascertained. For that purpose, choose any heavy body, forming part of the ship's lading, which can conveniently be shifted transversely.

Let P be the centre of gravity of that body in the upright position of the ship. Let the body now be shifted sideways through any convenient distance, PP', so that P' is the new position of its centre of gravity. The ship will heel over through an angle, which is to be accurately measured.

Let ZMZ' represent that angle of heel, so that MZ', in the new position of the vessel, represents a vertical line. Then the new position of the centre of gravity of the whole ship will be in the line, MZ'.

Through M draw MT parallel to PP', and less than PP' in the same proportion as that in which the shifted weight, P, is less than the whole weight of the vessel; that is, make—

$$\overline{MT} = \overline{PP'} \cdot \frac{\text{Shifted weight}}{\text{Displacement}}.$$

Through T draw TG' parallel to MZ, cutting MZ' in G'; this point will be the new position of the centre of gravity of the whole vessel. Through G' draw G'G parallel to P'P, cutting MZ in G; G will be the original centre of gravity of the whole vessel when upright, being the point which was to be found.

When PP', in the upright position of the vessel, is horizontal (as is usually the case), the following is the rule for finding the depth, MG, of the centre of gravity below the metacentre by calculation—

Multiply the shifted weight by the distance, PP', through which it is shifted; divide by the displacement; the quotient will be the distance GG' = MT through which the centre of gravity of the whole vessel is shifted; and this, multiplied by the cotangent of the angle of heel, gives the required depth, MG.

The displacement and the shifted weight should, of course, be expressed in the same units.

It may sometimes be convenient to shift several weights instead of one, and to shift them through different distances; and then each weight is to be multiplied by the distance through which it is

shifted, and the products added together; the sum so obtained is then to be used instead of the product of a single weight into the distance through which it is shifted.

EXAMPLE.—In a vessel of 800 tons displacement, materials weighing in all ten tons are shifted transversely through the following distances:—

5 tons shifted 14 feet,	Products.
5 tons shifted 12 feet,	70
	60
Sum,	130

The effect is to heel the vessel over to an angle of 2° 20'. What is the depth of the centre of gravity of the vessel below the metacentre?

$\frac{130}{800} = 0.1625$ foot, distance through which the centre of gravity of the vessel is shifted transversely.

Cotan. 2° 20' = 24.54.

$0.1625 \times 24.54 = 3.99$ feet, depth required.

A detailed account of several experiments of this kind, showing the precautions which must be observed in order to insure accuracy in their results, is contained in a paper by Mr. Barnes, published in the Transactions of the Institution of Naval Architects for 1860. Amongst those precautions the following may be specially mentioned; the hold should be pumped dry before the experiment; loose materials, such as coal, should be prevented from shifting; and the crew should be made to remain perfectly steady at their proper posts, while the inclination is being observed.

106. *Comparative Stability of Similar Vessels.*—If there be two or more vessels so shaped, that their load-water-lines differ only in their absolute dimensions of length and breadth, the ordinates or half breadths at the water-line in each vessel being proportional to the corresponding ordinates in each of the other vessels, it is evident, from the explanations in the preceding Articles, that the *moments of surface stability* in those vessels will be respectively proportional to the *lengths and the cubes of the breadths*.

The displacements of the same vessels will be proportional to the *lengths, the breadths, and the mean depths of immersion* (as to which last quantity, see Article 98); and hence the heights of the metacentres above the centres of buoyancy of the respective vessels will be proportional to the *squares of the breadths, divided by the mean depths of immersion*.

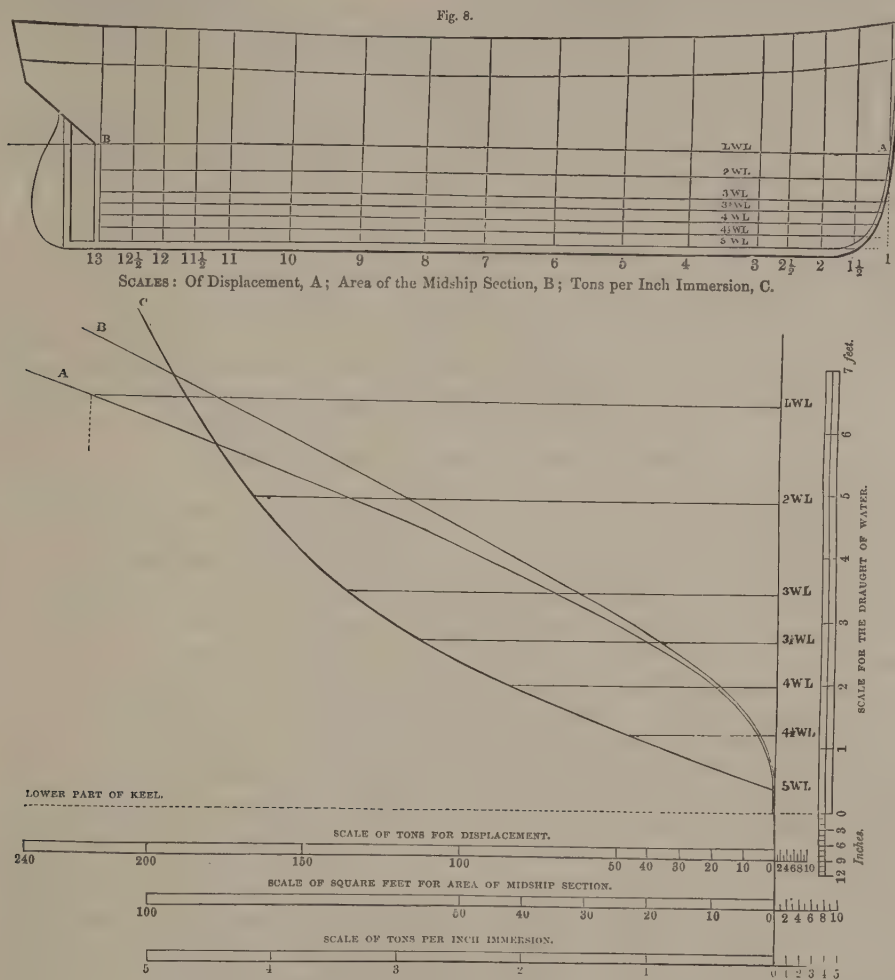
In practice, the mean depth of immersion is almost always greater as compared with the breadth in a large vessel than in a small one, and thus increases more rapidly than in the simple ratio of the breadth; indeed, in comparing some series of differently sized vessels together, the mean depth of immersion is found to increase almost, though not quite, in the proportion of the square of the breadth. The effect of this is, that the height of the metacentre above the centre of buoyancy, instead of increasing proportionally to the breadth (as it would do if the mean depth of immersion were proportional to the breadth simply), is found to increase much more slowly; and when with this fact is combined the proportionately greater height of large vessels out of the water, the result is that the height of the metacentre above the centre of gravity differs very little for well-designed vessels of all sizes (as already stated in Art. 101), being a little more or less than four feet, and in general greatest in the smallest vessels.

In vessels for which that height is exactly the same, the moments of stability at equal angles of heel are obviously proportional to the displacements.

SECTION III.—COMBINED CALCULATIONS OF BUOYANCY AND STABILITY.

107. The *Object of the present Section* is to illustrate by an example, the manner in which the calculations of displacement, and of the positions of the centre of buoyancy and metacentre, described in the two preceding Sections, can be conveniently combined in one tabular arrangement for practical purposes.

The methods of doing this are, of course, all identical in principle; but during the progress of naval architecture they have varied considerably in detail, and have been from time to time rendered more simple and concise. The arrangement adopted in this Section, as the most simple and concise yet known, is that which was devised by the late Mr. John Wilson, chief draughtsman in the surveyor's department of the Admiralty.



108. *Arrangement of the Data.*—The example chosen is an iron screw steamer, of which the following are the principal dimensions:—

Length on the load-water-line from the rabbet of the stem to the rabbet of the main stern-post (A B in Fig. 8),.....	Feet.	151.2
Extreme breadth,.....		20.0
Depth from upper part of beam to lower part of keel,.....		11.5
Draught of water, forward and aft (including keel),.....		6.5
The keel is 0.5 foot deep below the rabbet; therefore—Immersed depth from load-water-line to rabbet of keel,.....		6.0

The “appendages” of this vessel (Art. 94) are so small that they may be neglected.

On account of the smallness of the longitudinal scale, the vertical dimensions, for the sake of greater distinctness, are laid down to a scale about three times greater than the longitudinal dimensions; the longitudinal scale being about $\frac{1}{24}$ th of an inch to a foot, and the vertical scale about $\frac{1}{8}$ th of an inch to a foot. The sheer plan, Fig. 8, corresponds to the general description in

Art. 86. It shows that the length, A B, of 151.2 feet, is divided into 12 equal intervals of 12.6 feet, and that two of those intervals at the bow, and two at the stern, are subdivided into half-intervals. The cross sections are numbered from 1 to 13, commencing at the bow, those between the half-intervals having fractional numbers.

The ordinates or half breadths at the intersections of the cross sections and water-sections having been measured, are set down in the Table given at the end of this Article.

The column on the extreme left of that Table contains the numbers of the cross sections, 1, $1\frac{1}{2}$, 2, &c.

The next column contains Simpson's Multipliers, in their order, agreeably to the rules of Articles 19 and 24.

Then follow the columns containing the ordinates. Of these columns there are as many as there are water-sections; that is, in the present case, seven.

HYDRAULICS OF SHIPBUILDING.

The columns containing ordinates are headed at the top with the numbers of the water sections, and immediately below these with Simpson's Multipliers.

The ordinates are ranged in as many lines as there are cross sections; that is, in the present case, seventeen: thirteen being at whole intervals apart, and four at half-intervals.

No. of Cross Sections.	Simpson's Multipliers.	NOS. OF WATER-SECTIONS.												VERTICAL SECTIONS.				METACENTRE.			
		Keel or 5 W.L.		4½ W.L.		4 W.L.		3½ W.L.		3 W.L.		2 W.L.		L. W. L.		Half Areas ÷ V.L. 3	Multiples of Areas.	Multipliers for Leverage.	Moments.	Cubes.	Multiples of Cubes.
		1	2	1	2	1	2	1	2	1	2	1	2	1	2						
1	1	1 05	05	1 2	05	1 1	05	1 2	05	1 15	05	1 4	05	1 1	05	1 2	6	0	0	0	0
1½	2	1 05	2	1 2	2	2 2	4	3 6	6	4 6	8	7 28	14	9 9	18	5.35	10.7	½	5.35	73	1.46
2	1	1 05	1	2 4	2	4 4	4	6 12	6	8 12	8	13 52	13	19 19	19	10.35	10.35	1	10.35	6.86	6.86
2½	2	1 05	2	4 8	2	7 7	14	10 20	20	14 21	28	21 84	42	29 29	58	16.95	33.9	1½	50.85	24.39	48.78
3	1½	1 05	15	6 12	9	12 12	18	17 34	255	22 33	33	31 124	465	40 40	60	25.55	38.325	2	76.65	64.00	96.00
4	4	1 05	4	14 28	56	23 23	92	32 64	128	39 585	156	51 204	204	62 62	248	44.0	176.0	3	528.0	238.33	953.32
5	2	1 05	2	20 40	40	36 36	72	48 96	96	58 87	116	71 284	142	80 80	160	62.35	124.7	4	498.8	512.00	1024.00
6	4	1 05	4	26 52	104	48 48	192	63 126	252	75 1125	300	87 348	348	93 93	372	78.0	312.0	5	1560.0	804.36	3217.44
7	2	1 05	2	29 58	58	56 56	112	74 148	148	87 1305	174	96 384	192	100 100	200	87.7	175.4	6	1052.4	1000.00	2000.00
8	4	1 05	4	29 58	116	57 57	228	77 154	308	90 135	360	99 396	396	100 100	400	90.05	360.2	7	2521.4	1000.00	4000.00
9	2	1 05	2	26 52	52	50 50	100	68 136	136	80 120	160	92 368	184	97 97	194	82.35	164.7	8	1317.6	912.67	1825.34
10	4	1 05	4	18 36	72	35 35	140	49 98	196	60 90	240	77 308	308	87 87	348	65.45	261.8	9	2356.2	658.50	2634.00
11	1½	1 05	15	10 20	15	19 19	285	27 54	405	36 54	54	53 212	795	69 69	1035	42.85	64.275	10	642.75	328.50	492.75
11½	2	1 05	2	6 12	12	12 12	24	17 34	34	24 38	48	39 156	78	55 55	110	30.55	61.1	10½	641.55	166.38	332.76
12	1	1 05	1	3 6	3	6 6	6	10 20	10	14 21	14	25 100	25	38 38	38	19.15	19.15	11	210.65	54.87	54.87
12½	2	1 05	2	1 2	2	2 2	4	4 8	8	7 105	14	12 48	24	20 20	40	9.1	18.2	11½	209.3	8.00	16.00
13	1	1 05	05	1 2	05	1 1	05	1 2	05	1 15	05	1 4	05	1 1	05	1.2	6	12	7.2	.00	.00
Half Water-Sections, ÷ H.L. 3		3.6		55.2		103.95		141.5		171.4		209.7		236.95		V.L. 3 1832.0 = 5		H.L. 3 11689.05 = 12.6		H.L. 3 16703.68 = 4.2	
Simpson's Multipliers,.....		1		2		1		2		1½		4		1		H.L. 3 916.00 = 4.2		7013430 14026860		5340716 6681432	
Multiples of Water-Sections; sum 1832.00		1.8		110.4		103.95		283.0		257.1		838.8		236.95		1832 3664		1832) 147282.03		70155.036 2	
Multipliers for Leverage,....		4		3½		3		2½		2		1		0		3847.2 c. f.		Cr. of Buoyancy abait (1).		7694.4) 45770.024*	
Moments; sum 2765.95 V.L. =		7.2		386.4		311.85		707.5		514.2		838.8		0		2		7) 7694.4 c. f.		6.0784 ft.	
1382975 276595				18.0 441.6		1.8		165.6 311.85 141.5		236.95 838.8 171.4		1184.75 1677.6				5) 1099.2 Tons 219.84		236.95 4.2		Metacentre above Centre of Buoyancy.	
1832) 4148.925				459.6 103.95		164.175 V.L. = 5		622.55 H.L. = 4.2		V.L. = 1.5 H.L. = 4.2		V.L. = 5 H.L. = 4.2		2690.95 11301.99 V.L. = 1.5		Displacement to L. W. Line.		47390 94760 995.19 2		7) 1990.38 area of L. W. Section. 5) 284.34	
2.264 feet.				355.65 H.L. = 4.2		H.L. = 4.2		3922.065 3		2619.015 c. f. 2		6) 16952.985		3		V.I. 90.05 3 = 5		7) 1990.38 area of L. W. Section. 5) 284.34		12) 56.868 Tons per ft. Immersion. 4.739 Tons per in. Immersion.	
Cr. of Buoyancy below L. W. L.				533.475 3		344.7675 c. f.		8) 11766.195		7) 5238.03 c. f.		5) 748.20		5) 403.6425		45.025 sq. ft. 2		90.05 sq. ft.			
6.0784 2.264				12) 2240.595		7) 186.7162 c. f.		5) 98.505		5) 210.1106		149.658 tons. 219.84		80.7285 tons 219.84		Mid-Section to L. W. L.					
3.8144 feet.				5) 26.6737		5) 98.505		5) 210.1106		42.0221 tons.		70.182 tons.		139.1115 tons							
Metacentre above L. W. L.				5.3347 tons.		19.701 tons.		42.0221 tons.		70.182 tons.		139.1115 tons									
Dispt. to 4½ W.L.				Dispt. to 4 W.L.		Dispt. to 3½ W.L.		Dispt. to 3 W.L.		Dispt. to 2 W.L.											
Dispt. per In. at Lines,.....				1.104		2.079		2.830		3.428		4.194									
Mid-Section at Lines,.....				2.25		8.70		18.50		31.45		60.00									
																				* Coefficient of Surface Stability.	

109. *Arrangement of Results of Calculation.*—Immediately to the right of each ordinate is written, in differently-sized or differently-coloured figures, its product by the Simpson's multiplier proper to the line to which the ordinate belongs.

Immediately below each ordinate is written, in differently-sized or differently-coloured figures, its product by the Simpson's multiplier proper to the column to which the ordinate belongs.

For example, at the intersection of the line belonging to the cross section $1\frac{1}{2}$ (for which the Simpson's multiplier is 2), and the column belonging to the water-section 2 W.L. (for which the Simpson's multiplier is 4), is the ordinate .7. Immediately to the right of that ordinate is written its product by the multiplier 2; viz. 1.4; and immediately below it is written its product by the multiplier 4; viz., 2.8.

The products written below the ordinates are added in lines; and the sum of each line of products is written in the column headed "Half Areas + $\frac{V.I.}{3}$," under the general heading "Vertical Sections." The numbers in this column are proportional to the areas of the several vertical cross sections; but to give the absolute values of those areas, they still require to be multiplied by 2, and by one-third of the vertical interval of the ordinates, (abbreviated into $\frac{V.I.}{3}$.)

Each of those numbers proportional to the areas of the cross sections is then multiplied by the proper Simpson's multiplier, found in the second column from the extreme left of the Table; and the products are written in the column headed "Multiples of Areas." These multiples being added up, their sum (viz., 1832.0) is written at the foot of the column. It is then multiplied successively by one-third of the vertical interval ($\frac{V.I.}{3} = .5$) and by one-third of the horizontal interval ($\frac{H.I.}{3} = 4.2$). The product (3847.2) is one half of the load displacement, in cubic feet, which being multiplied by 2, gives 7694.4 cubic feet, the *whole load displacement*; and this, being divided successively by 7 and by 5, gives 219.84, the *Load Displacement in Tons*.

Each of the numbers in the column headed "Multiples of Areas" is next multiplied by the proper "Multiplier for Leverage," contained in the column on its right. The multiplier for leverage for a given cross section is the number of intervals by which that cross section is distant from the first cross section or commencement of the base-line (agreeably to the principles of Articles 38 and 96). The products are set down in the column headed "Moments;" and having been added up, their sum (11689.05, at foot of column) is multiplied by the horizontal interval (H.I. = 12.6). The product (147282.03) is not the absolute value of the moment of the displacement relatively to the first cross section; but it bears the same proportion to that moment which the sum of the column headed "Multiples of Areas" (1832) bears to the displacement. Dividing, therefore, that product by that sum, the quotient ($147282.03 \div 1832 = 80.29$) is the *horizontal distance in feet of the centre of buoyancy abaft the first cross section*.

Returning to the columns containing the ordinates, the products written immediately to the right of the ordinates are added in columns; and the sum of each column of products is written at the foot of the column, in the line marked "Half Water-Sections + $\frac{H.I.}{3}$." The numbers in this line are proportional to the areas

of the several water-sections; but to give the absolute values of those areas, they still require to be multiplied by 2, and by one-third of the horizontal interval between the ordinates (here abbreviated into $\frac{H.I.}{3}$).

Each of those numbers proportional to the areas of the water-sections is then multiplied by the proper Simpson's multiplier, as written in the line below it. The products are written in the next line again, marked "Multiples of Water-Sections" and being added together, their sum (1832.0) is written to their left. If the calculations have been correctly made, that sum ought to agree *exactly* with the sum of the column headed "Multiples of Areas."

Each of the numbers in the line of "Multiples of Water-Sections" is next multiplied by the proper "Multiplier for Leverage," contained in the line immediately below. The multiplier for leverage for a given water-section is the number of intervals by which that water-section is below the load-water-section (agreeably to the principles of Articles 38 and 96). The products are set down in the line marked "Moments;" and having been added together, their sum (2765.95, at the left end of the line) is multiplied by the vertical interval (V.I. = 1.5). The product (4148.925) is not the absolute value of the moment of the displacement relatively to the load-water-section; but it bears the same proportion to that moment which the sum of the line marked "Multiples of Water-Sections" (1832) bears to the displacement. Dividing, therefore, that product by that sum, the quotient ($4148.925 \div 1832 = 2.264$) is the *depth, in feet, of the centre of buoyancy below the load-water-section*.

Below the calculations of moments just described, are written the calculations of the *displacement up to the several water-sections* between the load-water-section and the keel. The calculator here employs various rules according to his judgment, so as to save labour as much as possible (see Article 93).

Thus, for the displacement up to 2 W.L., the volume of the layer between L.W.L. and 2 W.L. is computed by the rule of Art. 18A (cited in Art. 93), and subtracted from the load displacement.

For the displacement up to 3 W.L., the volume between L.W.L. and 3 W.L. is computed by Simpson's First Rule, and subtracted from the load displacement.

The displacement up to $3\frac{1}{2}$ W.L. is computed from the areas of the water-sections 5, $4\frac{1}{2}$, 4, and $3\frac{1}{2}$, by Simpson's Second Rule.

The displacement up to 4 W.L. is computed from the areas of the water-sections 5, $4\frac{1}{2}$, and 4, by Simpson's First Rule.

The displacement up to $4\frac{1}{2}$ W.L. is computed from the same areas by the rule of Article 18A.

The area of each water-section in square feet being divided by 7 and by 5, gives the *tons displacement per foot of immersion*, which is divided by 12 for the *tons displacement per inch of immersion*.

The *areas of the midship section* (No. 8) up to the several water-lines, are computed from its ordinates, just as the displacements are computed from the water-sections; and those areas are written at the foot of the Table.

The two columns at the right-hand side of the Table, headed "Metacentre," contain the *calculations of stability*, made according to the rule of Article 102.

The first of those columns, headed "Cubes," contains the cubes of the ordinates, or half breadths, of the load-water-section.

Each of those cubes is multiplied by the proper Simpson's multiplier (found in the second column from the left of the Table), and the products are written in the column headed "Multiples of Cubes." Those products having been added up, their sum (16703.58) is multiplied by one-third of the horizontal interval ($\frac{H.I.}{3} = 4.2$), giving the area of the curve whose ordinates are the cubes of the half breadths (70155.036). Two-thirds of that area is the *co-efficient of surface stability* (46770.024); which, being divided by the displacement in cubic feet, (7694.4), gives the *height of the metacentre above the centre of buoyancy* (6.0784 feet).

From that height, at the lower left-hand corner of the Table, is subtracted the depth of the centre of buoyancy below L.W.L. (2.264 feet); leaving the *height of the metacentre above L.W.L.* (3.8144 feet).

The scales of displacement, of tons per inch immersion, and of midship-areas, are shown in Fig. 9. (See Article 90.)

SECTION IV.—MORE EXACT CALCULATIONS OF STABILITY.

110. The *Objects* of the measurements and calculations described in this Section are as follows:—

I. To determine whether, and to what extent, the heeling of the vessel will be accompanied by rising or sinking of her centre of gravity.

II. To determine whether the heeling of the vessel will tend to produce pitching (that is, depression of the bow and raising of the stern), or scending (that is, depression of the stern and raising of the bow).

III. To find the moment of stability at finite angles of heel with greater exactness than is possible by means of the "metacentric" method described in Sections II. and III.

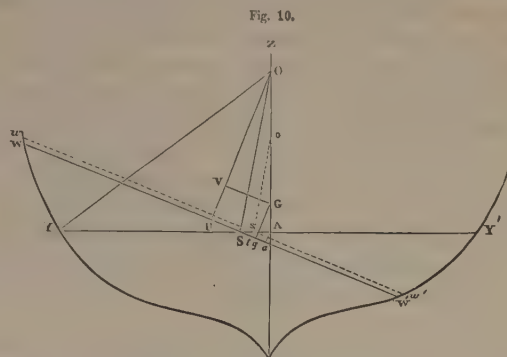
IV. To find the dynamical stability (a term introduced by the Reverend Canon Moseley to denote the work which is performed in heeling a vessel to a given angle).

The calculations comprehended under those four heads are made for one or more finite angles of heel, which are seldom much less than 7°, nor much more than 20°. When two or more angles of heel are adopted, it is advisable, for the sake of convenience in making the calculations, that they should form a series of multiples of one angle, such as 10°, 20°; or 7°, 14°, 21°; or 5°, 10°, 15°, 20°.

111. *True Position of Inclined Water-Section, and Volumes of Wedges of Immersion and Emersion.*—From the explanations already given in Article 100, it is evident, that in order to find accurately the position of a water-section inclined at a given angle to the upright water-section, it is necessary to find the true position of a plane which shall make the wedges of immersion and emersion equal. In most cases which occur in practice, it is impossible to solve this problem with perfect exactness by a direct process; but it is always possible to obtain a solution of any degree of precision that may be required by "trial and error;" that is, by a series of approximations, of which two are in general sufficient.

Let Fig. 10 represent an end view of a vessel; Y Y' her upright water-section; Z A her upright axis; G, her centre of gravity. The inclined water-section for any given angle of heel must cut the upright water-section either in the longitudinal axis, marked A, or in some straight line parallel to that axis. It is known that in all the forms of ships which commonly occur in practice, if the inclined water-section be supposed to traverse A, the wedge of

immersion is found to be greater than the wedge of emersion; whence it follows that the true line of intersection of the upright



and inclined water-sections must lie towards the immersed side of the vessel.

An approximate position, such as *s* in the figure, is assumed for that intersection, for the purpose of trial. In the British Navy it is usual to assume, for a first trial,

$$A s = .02 \text{ foot} \times \text{degrees in angle of heel.}$$

The assumed trial position of the inclined water-section, represented by the dotted line *ws w'*, cuts off a pair of trial wedges, *Y s w* and *Y' s w'*, whose contents are to be computed by the rule of Article 31. (An example of the application of that rule to this purpose will be given in Article 117.)

Should the volumes of those wedges prove to be equal, the assumed position of the inclined water-section is correct. Should they prove unequal, the following rules are to be applied:—

I. If the trial wedge of immersion is the greater, the assumed position is too high.

II. If the trial wedge of emersion is the greater, the assumed position is too low.

III. The perpendicular distance *st* of the corrected position from the assumed position, is found as follows:—Compute the area of the inclined water-section by the rules of Article 19, as applied in Article 92. Then

$$st = \frac{\text{Difference of Volumes of Wedges}}{\text{Area of Inclined Water-Section}}$$

IV. It is in general advisable to find also by calculation the distance *s S* from the trial position to the corrected position of the crossing of the inclined and upright water-sections, as follows:—

$$s S = \frac{st}{\sin. \text{angle of heel}}$$

Thus is found a corrected position, *W S W'*, of the inclined water-section; and in general, no further correction is required.

The corrected distance *A S* ($= A s + s S$), as thus found, ranges in ordinary cases from about 0.01 to 0.06 of a foot per degree of heel, its most common values being from 0.02 to 0.03 per degree.

112. *The Vertical Motion of the Centre of Gravity of the Vessel* produced by heeling through a given angle (if any) is found as follows:—In the upright position, the vertical distance of the centre of gravity from the plane of flotation is *GA*. In the inclined position, that distance becomes *Gg* (a perpendicular let fall from *G* to the inclined water-section *W S W'*).

$$\text{The difference of those distances,} \\ Gg - GA$$

is the vertical motion of the centre of gravity produced by heeling

through the given angle, and it is almost always an *elevation*; so that the centre of gravity is lifted when the vessel heels, and sinks again as she rights herself.

In the case of a *proposed* vessel, the centre of gravity is to a certain extent arbitrary; so that a probable position must be assumed for it, such as a certain distance (say four feet) below the metacentre. The centre of gravity is seldom far from the point, A, where the upright axis cuts the upright water-section; and hence an approximation to the extent of its vertical motion may be found by measuring the perpendicular, Aa , let fall from the point A upon the inclined water-section. The value of that perpendicular is

$$Aa = \overline{AS} \times \sin. \text{ angle of heel};$$

and it is obviously the less, the closer S is to A.

When the vertical motion of the centre of gravity is too great, it produces "uneasy" rolling, and other bad effects, which will be explained in Chapter IV. To keep that motion within moderate limits, it is desirable that the distance, AS, should not exceed about 0.02 or 0.03 of a foot per degree of angle of heel.

113. The term, *Axis of Level Motion*, may be applied to a horizontal line in the upright longitudinal plane ZA, so placed as to preserve the same level when the ship has heeled to a certain angle as it has when the ship is upright. The finding of the position of this line is a convenient step towards determining the vertical motion of the centre of gravity.

In Fig. 10, draw SO bisecting the angle WSY' between the upright and inclined water-sections; then O will be the point where the axis of level motion cuts the upright axis, ZA; because it is evident that the perpendicular distances, OU and OA, of that point from the inclined and upright water-sections, are equal to each other.

The common value of those perpendicular distances may be found by any one of the following rules:—

I. Find the distance \overline{AS} by the rules of Article 3; then

$$AO = \overline{AS} \times \cotan. \frac{1}{2} \text{ angle of heel.}$$

II. Find the perpendicular \overline{Aa} as in Article 112; then

$$\overline{AO} = \frac{\overline{Aa}}{\text{versin. angle of heel}}$$

III. Assume an approximate position, o , for the axis of level motion, and find the corresponding approximate intersection, s , of the upright and inclined water-sections, either by laying off the angle $Aos = \frac{1}{2}$ angle of heel, or by making $\overline{As} = \overline{Ao} \times \tan. \frac{1}{2}$ angle of heel. Then proceed, as in Article 3, to compute the distance, st , between the approximate and corrected positions of the water-section; when the correction of the position of the axis of level motion will be as follows;

$$\overline{oO} = \frac{\overline{st}}{\text{versin. angle of heel}}$$

IV. The following approximate method (founded on a theorem of Dupin's) will in most cases which occur in practice serve to determine the axis of level motion with sufficient accuracy, and will not involve the necessity for computing the volumes of the immersed and emersed wedges.

If the cross sections of the vessel between wind and water were all, like those represented in Fig. 6, Article 102, arcs of circles described about one longitudinal axis, O, that axis would evidently be itself the axis of level motion; and its position might be found by drawing YO perpendicular to the vessel's side at the water-line, to cut the upright axis in O.

When the cross section between wind and water in any interval of the length of a vessel consists of a pair of such circular arcs, and the axis of level motion of the vessel does not traverse the centre of these arcs, then the part of the wedge of $\left\{ \begin{array}{l} \text{immersion} \\ \text{emersion} \end{array} \right\}$ belonging to that interval will be in excess, according as the centre of the arcs is $\left\{ \begin{array}{l} \text{above} \\ \text{below} \end{array} \right\}$ the axis of level motion, and the excess for a given angle of heel will be proportional very nearly to the half breadth of the water-section, multiplied by the difference of level of the centre of the arcs and the axis of level motion. In a vessel whose cross sections between wind and water consist of circular arcs described about centres at different levels, the axis of level motion will lie between the highest and the lowest of those centres, and in such a position that the excesses of the immersed and emersed volumes, produced as above described, shall exactly neutralize each other.

The height above or depth below the water-line plane of the centre of the arcs at a given cross section, is equal to the half breadth, multiplied by the tangent of the angle made by the vessel's side at the water-line with the vertical; that centre being above or below water according as the side inclines outwards or inwards—in other words, "flares out" or "tumbles home."

Hence the following rule for calculation:—

Measure the angles of inclination of the several cross sections to the vertical between wind and water, and find their tangents, distinguishing those tangents into $\left\{ \begin{array}{l} \text{positive} \\ \text{negative} \end{array} \right\}$ according as the side

inclines $\left\{ \begin{array}{l} \text{outward} \\ \text{inward} \end{array} \right\}$. Multiply the tangents by the squares of the half breadths of the cross sections to which they respectively belong, and by Simpson's Multipliers in their proper order; take the algebraical sum of the products (that is, add together the positive products, and subtract the negative products). Multiply that sum by one-third of the longitudinal interval, and divide by the half area of the water-section (Art. 92); the quotient will be the height of the axis of level motion above the plane of flotation.

[In algebraical symbols, let dx be an indefinitely small interval of the ship's length; y the corresponding half breadth; c the inclination of the cross section between wind and water to the vertical, which is to be considered positive if outward, and negative if inward; then the height of the axis of level motion above the water-line is—

$$\overline{AO} = \frac{\int y^2 \tan c. dx}{\int y dx}.]$$

In a vessel whose cross sections at the water-line are vertical, and above and below the water-line are either straight and vertical, or equally and similarly curved, the axis of level motion exactly coincides with the longitudinal axis, A, of the water-section. Flaring, or outward inclination of the sides, tends to raise the axis of level motion: tumbling home, or inward inclination, to depress it. It is usual to build vessels with a considerable outward inclination of the sides near the bow and stern, where the breadths are small: the effect of this in raising the axis of level motion can be wholly or partially counteracted by means of a comparatively small inward inclination of the sides near the middle of the vessel's length, where the breadths are great.

V. The axis of level motion having been found, and being assumed (as it generally is) to be nearly the same for different angles of heel, the vertical motion of the centre of gravity for

any angle of heel may be found as follows: lay off the angle AOU equal to the given angle of heel; from G let fall GV perpendicular to OU ; then

$$\text{Vertical motion of centre of gravity} = \overline{OG} - \overline{OV} \\ = \overline{OG} \times \text{versin. angle of heel.}$$

For equal angles of heel, then, it appears that the vertical motion of the centre of gravity is proportional to its depth below the axis of level motion. So far, therefore, as easy rolling depends on smallness of the vertical motion of the centre of gravity, it is promoted by having the axis of level motion and the centre of gravity near together; and as the centre of gravity is usually not far from the water-line plane, it is accordingly found, that in vessels which roll easily, the height of the axis of level motion above the water-line usually ranges from one foot to three feet; and that when it much exceeds the latter limit, the rolling is uneasy. Even when the axis of level motion is very near the water-line-plane, the rolling becomes uneasy when the centre of gravity is much depressed; and hence it is sometimes necessary to raise the level of a heavy cargo by stowing below it a sufficient quantity of light materials called "*dunnage*."

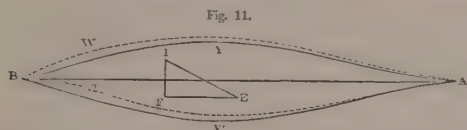
The following are a few examples of the height of the axis of level motion above the load-water-section—

Ship.	Authority for Data.	Angle of Heel.	Distance AS feet.	Height AO feet.
Vanguard,	Peake,	7°	0.43	6.94
Thetis,	Read,	7°	0.20	3.27
(Not named),	Barnes,	10°	0.23	2.63
Canopus,	Peake,	7°	0.08	1.30

114. Centres of Wedges—Pitching and Scending Moments.—

In order that the righting couple which is brought into action when a ship heels over to one side may tend simply to right the ship, and not to make her pitch or scend, by depressing the bow or stern as the case may be, it is necessary that the centres of the wedges of immersion and of emersion should be situated in one and the same plane of cross section, perpendicular to the axis of the ship.

Let Fig. 11 represent a horizontal or water-section of a ship, in which AB is the longitudinal axis, and $AYBY'A$ the upright



water-line; A being the bow, and B the stern. If the ship is of such a figure that when she heels over towards Y , the water-line becomes some such figure as that represented by the dotted line, $AWBW'A$, then the centre of the wedge of immersion, I , will be situated abaft the centre of the wedge of emersion, E . Through I draw IF perpendicular to AB ; and through E draw EF parallel to AB . Then the longitudinal distance, EF , by which the centre of the wedge of immersion is *abaft* the centre of the wedge of emersion, will be the arm of a couple of forces tending to raise the stern, and depress the bow—in other words, to make the ship *pitch*.

To exemplify the contrary case, we have only to suppose B to be the bow, and A the stern; when EF will become the distance by which the centre of the wedge of immersion is *afore* the centre of the wedge of emersion, and will be the arm of a couple tending to raise the bow and depress the stern, or to make the ship *scend*.

To determine the longitudinal positions of the centres of immersion and emersion, the moments of the wedges of immersion and emersion are to be taken relatively to a plane traversing A at right angles to the axis, AB , and divided by the volumes of those wedges, according to the method described in Article 40, which will be illustrated further on by a detailed example.

115. *Exact Statical Stability*.—According to the formula (II.) of Article 100, the moment of the righting couple for a given angle of heel is expressed as follows (see Fig. 12):—

Moment of Wedges—(Displacement $\times \overline{CR}$); the moment of the wedges of immersion and emersion being taken relatively to a plane traversing the intersection, S , of the upright and inclined water-sections, and perpendicular to the inclined water-section; and the line, \overline{CR} , being found by multiplying the height of the centre of gravity above the centre of buoyancy, by the sine of the angle of heel.

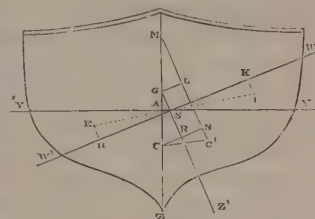
The approximate position of the intersection, S , having been assumed as already described in Article 111, the moment of the wedges is to be computed according to the rules of Article 40. A detailed example will be given in Article 117. It is seldom or never requisite to repeat the calculations with the corrected position of S , because sufficient accuracy can almost always be attained by means of a *correction*, computed in the following manner:—It is evident, on considering Fig. 10, that the error in the moment of stability occasioned by assuming the approximate position of the inclined water-section, $ws'w'$, instead of the true position, WSW' , consists simply of the moment of the layer contained between those two planes relatively to the assumed axis, s ; and is an error of $\left\{ \begin{smallmatrix} \text{excess} \\ \text{deficiency} \end{smallmatrix} \right\}$ according as the assumed water-section is $\left\{ \begin{smallmatrix} \text{too high} \\ \text{too low} \end{smallmatrix} \right\}$; that is, according as the wedge of

immersion, with the assumed water-section, is $\left\{ \begin{smallmatrix} \text{greater} \\ \text{less} \end{smallmatrix} \right\}$ than the wedge of emersion. Now the moment of the layer in question may be computed by taking the moment of the inclined water-section relatively to the assumed axis, s , according to the principles of Article 37; that is, the *difference* between the moments of the two half water-sections, because they are at opposite sides of that axis, and multiplying it by the thickness, st , of the layer, found as explained in Rule III. of Article 111. The correction thus calculated is to be $\left\{ \begin{smallmatrix} \text{subtracted} \\ \text{added} \end{smallmatrix} \right\}$ according as the assumed wedge of $\left\{ \begin{smallmatrix} \text{immersion} \\ \text{emersion} \end{smallmatrix} \right\}$ is the greater. This will be exemplified in Article 117.

In well-formed vessels of ordinary shapes, it seldom happens that the stability found by this more exact process differs much from that computed by the metacentric method. The example to be given will illustrate that fact. Hence it is chiefly when there is something unusual in the figure of the vessel between wind and water, that the more exact process of calculation is needed.

To find the *Shifting metacentre* for a given angle of heel, divide the moment of the wedges of immersion and emersion by the displacement, and by the sine of the angle of heel; the quotient

Fig. 12.



will be the height, MC , of the shifting metacentre above the centre of buoyancy. That shifting metacentre may be used in finding the centre of gravity by experiment, just as a fixed metacentre is used in Article 105.

116. The term *Dynamical Stability* has been applied by the Reverend Canon Moseley to the mechanical work which is performed in forcing a vessel to heel over to a given angle.* That work is performed, partly in raising the vessel's centre of gravity, and partly in depressing her centre of buoyancy; and it is computed by multiplying the vessel's displacement by the increase in the difference of level of her centres of gravity and of buoyancy. That is to say, its value is—

(I.) Displacement $\times (\overline{LC'} - \overline{GC})$ in Fig. 12; but $\overline{LN} = \overline{GC} \times \cos.$ angle of heel; consequently, the dynamical stability is—

(II.) Displacement $\times (\overline{NC'} - \overline{GC} \text{ versin. angle of heel})$; in which the term displacement $\times \overline{NC'}$ alone remains to be determined.

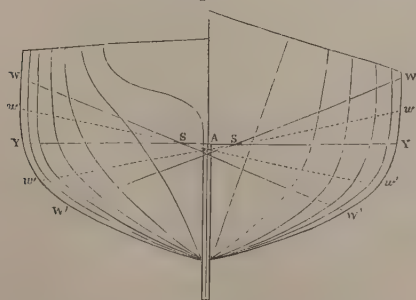
Now that term is equal to the difference between the moments, relatively to the inclined water-section, of the displacement in its original shape, $Y'ZY$, and in its new shape, $W'ZW$; and that difference is equal to the *arithmetical sum of the moments of the wedges of immersion and emersion relatively to the inclined water-section, $W'SW$* . The rule for computing such moments has been given in Article 40; and its application will be exemplified in Article 117.

An *approximation* to the dynamical stability at a given angle of heel is found by multiplying the displacement by the height of the metacentre above the centre of gravity, MG , and by the versed sine of the angle of heel.

Another method of approximating to the dynamical stability consists in multiplying the statical stability by the tangent of one-half of the angle of heel; because the righting moment varies nearly as the sine of the angle of heel, and the dynamical stability nearly as the versed sine of the same angle; and the versed sine of any angle is equal to its sine multiplied by the tangent of half the angle.

When the dynamical stability has been computed from an approximate position, $ws w'$ (Fig. 10) of the inclined water-section, its *correction* is found by multiplying the layer between the approximate and the true inclined water-sections (that is, the

Fig. 13.



difference between the wedges of immersion and of emersion) by half the thickness, st , of that layer; and that correction is to be {subtracted} according as the wedge of {immersion} is the greater.

117. *Method of Measurement, and Example of Calculations.*—Fig. 13 illustrates the method of preparing the body-plan of a ship for the measurements required in order to perform the calculations described in the preceding Articles of this Section. The right side of the plan shows the fore body; the left side, the after body. YY is the upright water-section. A pair of points, each marked S , indicate the assumed intersection of the inclined and upright water-sections. Through those points are drawn a pair of lines, each marked WSW' , to represent the inclined water-section on both fore and after bodies. The fore and after parts of the wedge of immersion are each marked WSY ; the fore and after parts of the wedge of emersion, $W'SY$. Intermediate sectional planes, radiating from S , are represented by the lines marked $ws w'$, which, in the example shown, bisect the angles, YSW , YSW' . One intermediate sectional plane is in general sufficient; so that the "angular interval" (Arts. 30, 40) is one half of the angle of heel, expressed in circular measure.

Six sets of ordinates or half breadths are then measured; viz.:—

FOR THE WEDGE OF IMMERSION.

- (1.) The ordinates of the upright water-section, from the nearer of the intersections S ;
- (2.) The ordinates of the intermediate section, $S w$; from the same point.
- (3.) The ordinates of the inclined water-section, $S W$; from the same point.

FOR THE WEDGE OF EMERSION.

- (4.) The ordinates of the upright water-section, from the farther of the intersections, S ;
- (5.) The ordinates of the intermediate section, $S w'$; from the same point.
- (6.) The ordinates of the inclined water-section, $S W'$; from the same point.

Upon each of those six sets of ordinates, the following operations are performed.

A. Treat their half squares as the ordinates of a new curve, and find its area by Simpson's First Rule. Let the results of this operation be called, $A_1, A_2, A_3, A_4, A_5, A_6$. Then,

$$\text{Vol. of wedge of immersion} = (A_1 + 4 A_2 + A_3) \times \frac{\text{angular interval}}{3}.$$

$$\text{Vol. of wedge of emersion} = (A_4 + 4 A_5 + A_6) \times \frac{\text{angular interval}}{3}.$$

As to the use of those volumes, see Articles 111, 112. Their difference, divided by the area of the inclined water-section, gives the thickness (st , Fig. 10) of the layer between the assumed and the true positions of that section. The area of the inclined water-section is computed as in Article 92; observing, however, that it consists in general of two unequal divisions.

B. Multiply each of the before-mentioned half squares by the longitudinal distance of the ordinate from the foremost cross-section; treat the products as the ordinates of a new curve and find its area; let the results of this operation be called, $B_1, B_2, B_3, B_4, B_5, B_6$. Then, moments relatively to the foremost cross-section

$$\text{Of the wedge of immersion} = (B_1 + 4 B_2 + B_3) \times \frac{\text{angular interval}}{3}.$$

$$\text{Of the wedge of emersion} = (B_4 + 4 B_5 + B_6) \times \frac{\text{angular interval}}{3}.$$

As to the use of those moments, see Article 114.

C. Treat the third parts of the cubes of the ordinates as the ordinates of a new curve, and find its area; let the results of this operation be called $C_1, C_2, C_3, C_4, C_5, C_6$. Then,

Moment of statical surface-stability

$$= \{C_3 + C_5 + 4(C_2 + C_4) \cos. \frac{1}{2} \text{ angle of heel} \\ + (C_1 + C_6) \cos. \text{ angle of heel}\} \times \frac{\text{angular interval}}{3};$$

INCLINED WATER SECTION.

IMMERSED WEDGE.												EMERGED WEDGE.															
Number of Ordinates.	Ordinates.	Simpson's Multipliers.	Functions of the Ordinates.	Squares of Ordinates.	Simpson's Multipliers.	Functions of the Squares.	Intervals from (1).	Longitudinal Moments.	Cubes of the Ordinates.	Simpson's Multipliers.	Functions of the Cubes.	Number of Ordinates.	Ordinates.	Simpson's Multipliers.	Functions of the Ordinates.	Squares of Ordinates.	Simpson's Multipliers.	Functions of the Squares.	Intervals from (1).	Longitudinal Moments.	Cubes of the Ordinates.	Simpson's Multipliers.	Functions of the Cubes.				
1	0	1	0	00	1	00	0	00	00	1	00	1	0	1	0	00	1	00	0	00	00	1	00				
1½	7	2	1.4	49	2	98	1	49	34	2	68	1½	1.1	2	2.2	1.21	2	2.42	1	1.41	1.33	2	2.66				
2	1.8	1	1.8	3.24	1	3.24	1	3.24	5.83	1	5.83	2	2.0	1	2.0	4.00	1	4.00	1	4.00	8.00	1	8.00				
2½	3.0	2	6.0	9.0	2	18.00	1½	27.00	27.00	2	54.00	2½	2.9	2	5.8	8.41	2	16.82	1½	25.23	24.39	2	48.78				
3	4.3	1	6.45	18.49	1	27.74	2	55.48	79.51	1½	119.26	3	3.0	1½	5.85	15.21	1½	22.81	2	45.62	59.32	1½	88.98				
4	6.7	4	26.8	44.89	4	179.56	3	538.68	300.76	4	1203.04	4	5.8	4	23.2	33.64	4	134.56	3	403.68	195.11	4	780.44				
5	8.4	2	16.8	70.56	2	141.12	4	564.40	592.70	2	1185.40	5	7.6	2	15.2	57.76	2	115.52	4	462.08	438.98	2	877.96				
6	9.5	4	38.0	90.25	4	361.00	5	1805.00	857.38	4	3429.52	6	9.0	4	36.0	81.00	4	324.00	5	1620.00	729.00	4	2916.00				
7	10.0	2	20.0	100.00	2	200.00	6	1200.00	1000.00	2	2000.00	7	9.9	2	19.8	98.01	2	196.02	6	1176.12	970.30	2	1940.60				
8	10.0	4	40.0	100.00	4	400.00	7	2800.00	1000.00	4	4000.00	8	10.2	4	40.8	104.04	4	416.16	7	2913.12	1061.21	4	4244.84				
9	9.7	2	19.4	94.09	2	188.18	8	1505.44	912.67	2	1825.34	9	9.5	2	19.0	90.25	2	180.50	8	1444.00	857.38	2	1714.76				
10	9.0	4	36.0	81.00	4	324.00	9	2916.00	729.00	4	2916.00	10	8.2	4	32.8	67.24	4	268.96	9	2420.64	551.37	4	2205.48				
11	7.8	1	11.7	60.84	1	91.26	10	912.60	474.55	1	711.82	11	6.1	1	9.15	37.21	1	55.82	10	558.20	226.98	1	340.47				
11½	6.5	2	13.0	42.25	2	84.50	10½	887.25	274.63	2	549.26	11½	4.9	2	9.8	24.01	2	48.02	10½	504.21	117.65	2	235.30				
12	4.6	1	4.6	21.16	1	21.16	11	232.76	97.34	1	97.34	12	3.5	1	3.5	12.25	1	12.25	11	134.75	42.88	1	42.88				
12½	2.2	2	4.4	4.84	2	9.68	11½	111.82	10.65	2	21.30	12½	2.0	2	4.0	4.00	2	8.00	11½	92.00	8.00	2	16.00				
13	0	1	0	00	1	00	12	00	00	1	00	13	0	1	0	00	1	00	12	00	00	1	00				
				246.35					1855.96									229.4					1805.96				
				229.40					1805.96													1805.96					
				475.75					2050.42													1805.96					
				1998.15					1805.96													1805.96					
				1998.15					1805.96													1805.96					
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Explanation of the Table of Calculations of the Surface Stability of a Ship.—The example given is of a ship 150 feet long, 20 feet broad, and of 310 tons burthen, builder's measure, being the same vessel which has already been used as an example in Section III. The displacement is 7694·4 cubic feet, or 219·84 tons. The point S is assumed to be two-tenths of a foot from the middle line when the inclination of the ship is 10° from the upright position. The angles of the wedges of immersion and emersion are each divided into two equal intervals. The Table is consequently divided into three parts corresponding to the three water-sections, and each of those into two parts for the immersed and emerged wedges respectively. In the Table the immersed wedge is to the left hand, and the emerged wedge to the right hand. By an inspection of the Table it will be seen that the columns for the upright water-section and intermediate section are precisely the same, and those for the inclined water-section differ only in there being additional columns for obtaining the area of that section, a result not required for the other sections.

The length of the ship at the load-water-line is divided into twelve intervals of 12·6 feet; and four of those, near the end, are subdivided into half-intervals.

The columns marked "Number of Ordinates" and "Ordinates" are explained by their headings. The column headed "Squares of Ordinates" is obtained from a table of the squares of numbers, each square number being placed exactly opposite to its corresponding ordinate. The column headed "Functions of the Squares" contains the products of the squares of the several ordinates, and of the numbers standing opposite to them in the column marked "Simpson's Multipliers." The sum of the Functions of the Squares when multiplied by $\frac{1}{3}$, and by one-third of the longitudinal interval, is equal to the moment of the plane to which it belongs, relatively to S. The above multiplications are, however, not performed; but the sum of the quantities in each of those columns for the immersed and emerged wedges is considered as the ordinate of a curve; the sums for the inclined and upright water-sections are each multiplied by 1, and those for the intermediate section by 4, and when the three products for each of the wedges have been added together, the results are each multiplied by $\frac{1}{3}$, by 4·2 (one-third of the longitudinal interval), and again by ·02909 (one-third of the angular interval), and the volumes of the wedges of immersion and emersion are obtained. The columns for obtaining the longitudinal position of the centres of gravity of the immersion and emersion are headed "Longitudinal Moments." The "Cubes of the Ordinates" are obtained from a table of the cubes of numbers, and are inserted in the column opposite to the respective ordinates. The column headed "Functions of the Cubes" contains the products of the cubes of the several ordinates and the numbers standing opposite to them in the column "Simpson's Multipliers." The sum of the Functions of the Cubes, when multiplied by $\frac{1}{3}$ and by 4·2, is equal to the moment of the inertia of the plane section to which it belongs. The above multiplications are, however, not performed; but the two sums of the quantities in those columns for each section are added together, the results for the upright and inclined water-sections are each multiplied by 1, and that for the intermediate section by 4. In obtaining the statical surface stability, the above product for the inclined water-section is multiplied by $\cos. 0^\circ$ (or 1), that for the intermediate section by $\cos. 5^\circ$, and that for the upright water-section by $\cos. 10^\circ$; those three quantities added together, and the result multi-

plied by $\frac{1}{3}$, by 4·2, and by ·02909, give the statical surface stability at 10° angle of heel.

To obtain the dynamical surface stability, the quantities which were multiplied by $\cos. 0^\circ$, $\cos. 5^\circ$, and $\cos. 10^\circ$ are respectively multiplied by $\sin. 0$ (or 0), $\sin. 5^\circ$, and $\sin. 10^\circ$, and their sum multiplied by the same factors as before; the result gives the value of the dynamical surface stability at 10° angle of heel.

In the example given the volume of immersion is found, on calculation, to exceed that of emersion by 10·4255 cubic feet, which shows that the position assumed for the intersection S of the upright and inclined water-sections is only approximate, and not exact. The corrections which have consequently to be subtracted from the stabilities, are given at the end of the Table.

The last calculation is a comparison between the approximate statical stability, as calculated by the metacentric method, and the exact statical stability; and it shows that in the example chosen, the error of the metacentric method is only $\frac{1}{85}$ th of the whole stability.

The correction in the position of the intersection S (which is to be added to its assumed distance from the longitudinal axis, A, of the vessel, is found by Rule IV. of Art. 111, as follows—

$$\frac{\text{Thickness of Layer, } st}{\sin. \text{ angle of heel}} = \frac{·00522}{·17365} = ·03 \text{ foot;}$$

consequently the true distance of S from A is, as already stated in Article 113—

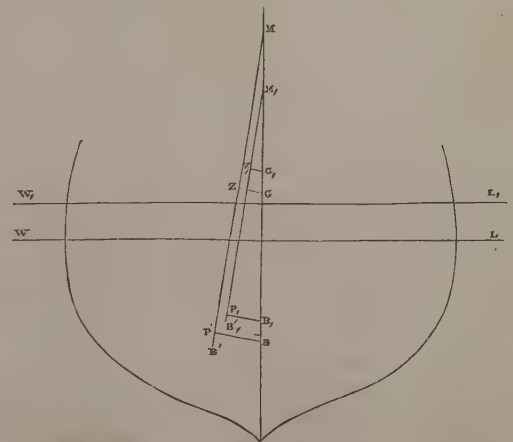
$$·2 + ·03 \times ·23 \text{ foot.}$$

117A. Effects of Addition or Removal of Weights on Stability.—

The effect of any great alteration in a ship's lading upon her stability may be found by first finding her altered depth of immersion, by the aid of the scale of displacement, and calculating the stability corresponding to the new plane of flotation by the rules already given. But for moderate alterations of the lading it is unnecessary to go through so laborious a process, as there is an easier method of finding the alteration of stability with sufficient accuracy.

Since the positions of the centres of gravity of ships are in general but roughly approximated to, it becomes important to be able to dispense with the knowledge of the exact position of the

Fig. 13A.



centre of gravity of a ship, and to compare the stability under given conditions with the stability under other conditions; the difference of the two cases being known. In Fig. 13A—

Let $W L$ represent the water-line of a ship.

G , her centre of gravity.

B , her centre of buoyancy.

Let the ship be now inclined about the fore and aft horizontal axis, through an angle, I . Let B' be her new centre of buoyancy.

Through B' draw $B'M$ vertical, and cutting the original vertical through B in the point M . Also through B and G draw $B P$ and $G Z$ respectively perpendicular to $B'M$.

Then if W represent the displacement of the ship—

$$\begin{aligned}\text{Stability} &= W \times G Z \\ &= W \times G M \sin. I \\ &= W \times (B M - B G) \sin. I \\ &= W \cdot B M \sin. I - W \cdot B G \sin. I.\end{aligned}$$

Now $W \cdot B M \sin. I = W \cdot B P = \bar{b} \cdot A$, where A represents the volume of either of the wedges of immersion or emersion, and \bar{b} the distance between the centres of gravity of these wedges.

Suppose the ship to be once more upright; and let a weight, w (the same unit being taken as for the displacement, W), be placed on board of her, causing her to sink in the water until $W_1 L_1$ becomes the new water-line.

Let the distance of the common centre of gravity of the weight or weights w added, above the original centre of buoyancy, be represented by a .

Let also c represent the distance of the centre of gravity of the additional displacement, above the original centre of buoyancy.

Let B_1 , and G_1 , represent the new centres of buoyancy and gravity respectively of the ship.

Let the ship be now inclined as before about a fore and aft axis, through the angle I . The centre of buoyancy is found at a point B'_1 .

Through B'_1 draw $B'_1 M_1$ vertical, cutting the original vertical $B G M$ in M_1 ; and through B_1 and G_1 draw $B_1 P_1$ and $G_1 Z_1$, respectively perpendicular to $B'_1 M_1$.

The stability is now represented by—

$$\begin{aligned}(W + w) G_1 Z_1 &= (W + w) G_1 M_1 \sin. I \\ &= (W + w) (B_1 M_1 - B_1 G_1) \sin. I \\ &= (W + w) B_1 M_1 \sin. I - (W + w) B_1 G_1 \sin. I. (A.)\end{aligned}$$

But $(W + w) (B_1 M_1 \sin. I) = (W + w) B_1 P_1 = \bar{b}_1 A_1$, where A_1 represents the volume of either of the wedges of immersion or emersion corresponding to the water-line $W_1 L_1$, and \bar{b}_1 represents the distance between the centres of gravity of those wedges.

Again $B_1 G_1 = B G + G G_1 - B B_1$;

Therefore $(W + w) B_1 G_1 \sin. I = W \cdot B G \sin. I + W \cdot B G \sin. I + (W + w) G G_1 \sin. I - (W + w) B B_1 \sin. I$.

But $w \cdot B G \sin. I + (W + w) G G_1 \sin. I =$ the moment of the weights added (w) about the original centre of buoyancy $= w a$.

Also $(W + w) B B_1 \sin. I =$ the moment of the additional displacement about the original centre of buoyancy $= w c$.

Substituting these values for their respective quantities in equation (A); the stability of the ship at the inclination I , corresponding to the water-line $W_1 L_1$ —

$$= \bar{b}_1 A_1 - W \cdot B G \sin. I - w (a - c) \sin. I. (B.)$$

The stability corresponding to the water-line $W L$, when the inclination of the ship is I from the upright position—

$$= \bar{b} A - W \cdot B G \sin. I. (C.)$$

subtracting equation (C) from (B), it is found that the difference of the stabilities of the ship under the conditions above stated—

$$= \bar{b}_1 A_1 - \bar{b} A - w (a - c) \sin. I. (D.)$$

Ships are commonly of such a form in the neighbourhood

of the load-water-line, that with a slight increase in the draught of water, the form of the new load-water-section is nearly the same as that of the original load-water-section.

Also if the ship be inclined, about a fore and aft axis, through the same angle, the wedges of immersion respectively corresponding to the two draughts of water will be very nearly the same; that is, see equation (D)—

$$\bar{b}_1 A_1 \text{ is practically equal to } \bar{b} A;$$

and the difference in the stability of the ship at the two draughts of water becomes $= -w (a - c) \sin. I$.

Consequently if the centre of gravity of the weights added be situated in the same horizontal plane as the centre of gravity of the additional displacement (that contained between the water-lines $W_1 L_1$ and $W L$), in which case $a = c$, the stability of the ship will be the same at both draughts of water.

If the centre of gravity of the weights added be above the centre of gravity of the additional displacement, i.e., if a be greater than c , the stability will be diminished by the quantity $w (a - c) \sin. I$. If the centre of gravity of the weights added be below that of the additional displacement, in which case c is greater than a , the stability is increased by the quantity $w (c - a) \sin. I$.

If $c - a = \bar{G} \bar{M}$, the height of the metacentre above the centre of gravity will not be altered by the additional lading.

In a ship floating at a given draught of water, if some of her weights have to be removed; the alteration in the stability by the removal of the weights can also be ascertained in the same manner.

Let $W_1 L_1$ represent the deep water-line, and $W + w$ the corresponding displacement.

Let w be the weights to be taken out. Then using the same notation as before: stability before the weights were removed

$$= \bar{b}_1 A_1 - W \cdot B G \sin. I - w (a - c) \sin. I.$$

$$\begin{aligned}\text{Stability when the weights are removed—} \\ &= \bar{b} A - W \cdot B G \sin. I.\end{aligned}$$

Therefore the stability in the second case minus the stability in the first case

$$= \bar{b} A - \bar{b}_1 A_1 + w (a - c) \sin. I.$$

If the ship be of such a form as to give—

$$\bar{b} A = \bar{b}_1 A_1$$

Then $w (a - c) \sin. I$ will represent the increase in the stability by the removal of the weights. That is—

When a is greater than c , or the centre of gravity of the weights removed is situated above the centre of gravity of the diminished displacement (between $W_1 L_1$ and $W L$) the stability of the ship is increased by the removal of the weights.

When the centre of gravity of the weights removed is situated in the same horizontal plane as that of the diminished displacement, the stability is the same as it was before the weights were removed.

When the centre of gravity of the weights removed is situated below that of the diminished displacement, in which case c is greater than a , the stability of the ship is diminished, by the removal of the weights, w , by the quantity $w (c - a) \sin. I$. If $c - a = \bar{G} \bar{M}$, the height of the metacentre above the centre of gravity is not altered by the removal of the weights.

It frequently happens, however, that the additional weights put on board, or taken out of, a ship are considerable, and they cause a sensible alteration in the form of the load-water-section, and consequently in the form of the wedges of immersion and emersion. In order to compare the stability of the ship in her original

condition, with that corresponding to the altered condition, it becomes necessary to calculate the wedges of immersion and emersion corresponding to the two water-lines WL and W_1L_1 (Fig. 13A), and their moments about the lines at their junction, by the process already described.

These calculations, however, are not rapidly performed; and since the relation between the stability of the ship in the two conditions only is required, the use of the following easy method will be attended with considerable advantages, and will be sufficiently exact for practical purposes.

In ships of a common form the ratio of the stability of one ship to that of another will be always substantially the same for ordinary equal inclinations, and nearly the same as the ratio of their stabilities when the inclination is evanescent. In the latter case the points in which the verticals through the centres of buoyancy corresponding to the inclined positions cut the original verticals through the centres of buoyancy corresponding to the upright position, are the metacentres.

Referring to Fig. 13A, suppose the angle I to be exceedingly small; then the points M and M_1 are the metacentres corresponding to the water-lines WL and W_1L_1 respectively.

Using the same notation as before, and taking the case where the weights w (measured in cubic feet of sea-water) are put on board of the ship whose displacement in cubic feet of sea-water is W :—

The stability corresponding to the water-line WL
 $= W (BM - BG) \sin. I$

and the stability corresponding to the water-line W_1L_1 is—
 $= (W + w) (B_1M_1 - B_1G_1) \sin. I,$

and these may be represented relatively by $W.BM - W.BG$ and $(W + w) B_1M_1 - (W + w) B_1G_1$ respectively.

Now $W.BM$ may be obtained from the Table used for calculating the height of the metacentre corresponding to the water-line WL . Let this quantity be represented by M .

Also $(W + w) B_1M_1$ may be taken from the table in which was calculated the height of the metacentre corresponding to the water-line W_1L_1 . Let this quantity be represented by M_1 ; then as before:

$$B_1G_1 = BG + GG_1 - BG_1,$$

and $(W + w) B_1G_1 = W.BG + w.BG + (W + w) GG_1 - (W + w) BG_1,$
 $BB_1 = W.BG + w(a - c),$

where a is the distance of the centre of gravity of the additional weights w put on board above the original centre of buoyancy; and c is the distance of the centre of gravity of the displacement between the water-lines WL and W_1L_1 above the original centre of buoyancy.

The stability of the ship in the two conditions may therefore be represented by—

$$M - W.BG,$$

and $M_1 - W.BG - w(a - c)$ respectively,

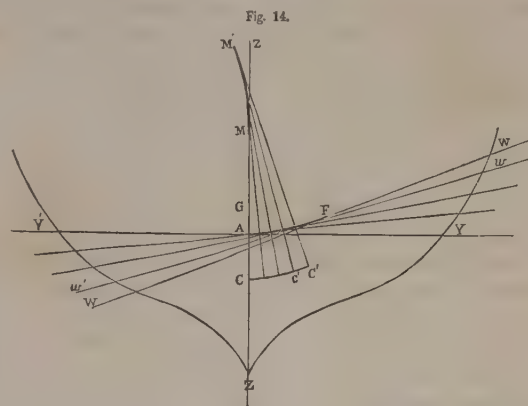
and the alteration in the stability, by the addition of the weights, may be represented by $M_1 - M - w(a - c)$.

And if the position of G be known or assumed, the proportionate loss or gain of stability will be readily found.

118. *Surface of Flotation—Metacentric Surface, Involute and Evolute.* This Article relates mainly to certain mathematical principles which, although intimately connected with the principles and rules explained in this chapter, are not usually applied directly to practice.

Suppose an unlimited number of water-sections (or *planes of flotation*, as they are sometimes called) to traverse a vessel in,

different positions, making indefinitely small angles with each other, and that a curved surface touches all those planes (being what mathematicians call their *envelope*): that curved surface is called the "SURFACE OF FLOTATION." For example, in the vertical section, Fig. 14, ZZ is the upright axis of the



vessel, G her centre of gravity, YY the upright water-section, and WW an inclined water-section; the straight lines between YY and WW represent an indefinite number of water-sections at intermediate angles of inclination, differing from each other by indefinitely small angular intervals; and the curve AF , which touches all those lines, represents a vertical section of part of the surface of flotation. The most important property of the surface of flotation is the following :—

I. *The point of contact of the surface of flotation with any water-section is the centre of that water-section.* (The word "centre" is to be understood as explained in Article 33.) This proposition is proved as follows: Let WW' and ww' be a pair of water-sections so close to each other in position, that their points of contact with the surface of flotation may be treated as if they coincided with each other at F . That pair of water-sections will intersect each other in a straight line, which will pass at an indefinitely small distance from F , and may be treated as if it traversed F . The pair of water-sections will contain between them one pair of indefinitely thin wedges, viz., a wedge of immersion WFw , and a wedge of emersion $W'Fw'$; and the volumes of those wedges must necessarily be equal to each other. Now in Article 31 it has been explained, that the volume of a wedge having a given angle at its edge is proportional to the moment of the longitudinal section of that wedge relatively to its edge; therefore the moments of the two divisions, FW and Fw' , into which the water-section WW' is divided by the before-mentioned straight line traversing F (being the edge of the two wedges), are equal to each other; therefore that line traverses the centre of the whole water-section WW' . But any straight line traversing the point of contact, F , may be indefinitely near to the line of intersection of a pair of water-sections; therefore the point of contact, F , must itself be the centre of the water-section WW' .

The centre of a given water-section is called a *centre of flotation*.^{*}

* The term "Centre of Flotation" is often used by naval architects to denote the middle of the length of the load-water-line, as measured between the rabbets of the stem and of the stern-post. That point, however unlike the centre of the plane of flotation, has no special mechanical properties, and its position is practically unimportant; it has therefore been considered preferable to use the term "Centre of Flotation" in the sense explained in the text.

When the vertical transverse section of the surface of flotation is a circle, the centre of that circle is in the *axis of level motion* of Article 113.

A surface traversing the indefinitely numerous positions of the centre of buoyancy which correspond to the indefinitely numerous water-sections, is called the *Metacentric Surface*. A transverse vertical section of the metacentric surface, such as that represented by the curve CC' in Fig. 11, is sometimes called the *Metacentric Involute*, for a reason which will be stated further on. The metacentric surface has the following property:—

II. *The metacentric surface at any given point, is parallel to the water-section for which that point is the centre of buoyancy.* This proposition is proved as follows: Let C' , c' , be the indefinitely close centres of buoyancy corresponding to the indefinitely close water-sections WW' , ww' . Then (as has been proved in Art. 100), the line $c'C'$ is parallel to a straight line joining the respective centres of the wedges of immersion and emersion, WFw and $W'Fw'$. But that line makes an indefinitely small angle with the water-section WW' ; therefore the metacentric surface at C' is parallel to the water-section WW' which corresponds to C' as a centre of buoyancy.

The statical and dynamical stabilities of the vessel are the same with those of a block of the same weight, having its centre of gravity, G , in the same position, and having a convex base of the figure of the metacentric surface, CC' , resting on a level platform (as represented in Fig. 4 of Chapter I., Art. 4); but the motions of the centres of gravity of the vessel and the block are not the same, except thus far:—the *relative motion* of the centre of gravity and point of support of the block, is the same with the *relative motion* of the centre of gravity and centre of buoyancy of the vessel. The following proposition is therefore evident:—

III. *Positions of equilibrium correspond to centres of buoyancy at which a straight line from the centre of gravity of the vessel is normal to the metacentric surface; and the equilibrium is stable when the length of that line is a minimum, and unstable when it is a maximum.*

From each of the centres of buoyancy contained in the metacentric involute, or vertical transverse section CC' of the metacentric surface, let a line be drawn (such as CM , $C'M'$, and the intermediate lines) normal to that surface; each of those lines will be normal also (by Prop. II. of this Article) to the corresponding water-section, and will represent the direction of the upward resultant pressure of the water through the centre of buoyancy which it traverses. The “envelope” of all those lines—that is, a curve, MM' , which touches them all, is called the “*Metacentric Evolute*,” being the curve from which, if a string be unwrapped with a pencil attached to it, starting from the upright centre of buoyancy, C , that pencil will describe the metacentric involute, CC' . Each of those straight lines touching the metacentric evolute, such as CM and $C'M'$, is the radius of curvature of the metacentric involute at the point (or centre of buoyancy) where it meets that curve, and may therefore be called a *Metacentric Radius*; and the point where it touches the metacentric evolute is the corresponding centre of curvature of the metacentric involute. The following property is possessed by the metacentric radii.

IV. *The Metacentric Radius, at a given Centre of Buoyancy, is equal to the Moment of Inertia of the corresponding water-*

section, divided by the displacement. This is proved as follows: Let WW' be any water-section; ww' , another water-section indefinitely close to it; C' , the centre of buoyancy for WW' ; c' , the centre of buoyancy for ww' ; and let the indefinitely small angular interval, in circular measure, between those water-sections be denoted by $d\theta$. Then by the reasoning already given in Article 100, it appears, that

$$\overline{C'C'} = \frac{\text{Moments of Wedges } W F w \text{ and } W' F w'}{\text{Displacement of vessel}};$$

but from Art. 103, it appears, that the moment of an indefinitely thin wedge is equal to the moment of inertia of its longitudinal section, multiplied by the indefinitely small angle at its edge, in circular measure. Therefore,

$$\overline{C'C'} = \frac{\text{Moment of Inertia of Water-section } WW' \times d\theta}{\text{Displacement}}.$$

Now the radius of curvature of a given arc of a curve is equal to the length of that arc divided by the difference of direction of its ends, in circular measure; and the difference of direction of the ends of the arc $\overline{C'C'}$ is $d\theta$ (being equal to that of the water-sections, WW' and ww' , by Prop. II. of this article); and therefore,

$$\text{Meta. Radius } \overline{C'M'} = \frac{\overline{C'C'}}{d\theta} = \frac{\text{Moment of Inertia of Water-section } WW'}{\text{Displacement}}.$$

[In symbols, let dx denote an indefinitely short interval of the length of the vessel; y' and y'' , the two half breadths or ordinates of the water-section WW' corresponding to that interval; then

$$C'M' = \frac{\frac{1}{3} \int (y'^3 + y''^3) dx}{\text{Displacement}}.]$$

It is evident that the point M , being the centre of curvature of the metacentric involute at the centre of buoyancy, C , corresponding to the upright water-section, YY , is identical with the *metacentre* for indefinitely small angles of heel, whose nature and use have already been explained in Article 103.

In vessels of the figure described in Art. 102, the metacentric evolute becomes a point, and the involute an arc of a circle.

The preceding propositions might have been made the foundation of the whole theory of the stability of ships, and of the rules for computing stability; but as it is difficult for a reader to feel the force of the reasoning by which they are proved unless he is familiar with the application of the differential calculus to the geometry of curved surfaces, it has been judged advisable to set out from principles of a more elementary kind.

SECTION V.—LONGITUDINAL STABILITY.

119. *The Determination of the Centre of Flotation, or Centre of the Upright Water-section*, is a process which it has not been necessary to consider in connection with transverse stability; because that centre is necessarily in the longitudinal axis, which divides that water-section into two halves symmetrical to each other. If it were for any reason desirable to design a ship with dissimilar sides, it would then be necessary to determine by measurement and calculation, the transverse position of the centre of flotation.

In Article 118, it has been shown that every small angular displacement of a ship from the upright position takes place about an axis traversing the centre of flotation. Hence it appears that pitching and scending, which are angular displacements in a longitudinal plane, take place about a transverse axis passing through the centre of flotation; and therefore the determination of

the position of that centre is the first step towards finding the longitudinal stability.

When the forward division and aft division of the load-water-line are of equal length and similar figure, the centre of flotation is at the middle of the length of the load-water-line.

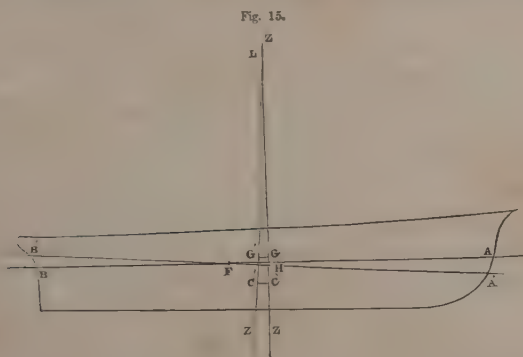
When the whole series of water-lines at different immersions are curves of the same kind, and have their ordinates in the same vertical section proportional to each other, the centre of flotation is in the upright axis passing through the centre of buoyancy.

In all other cases, the distance of the centre of flotation from ordinate No. 1, is to be found, according to Rule II. of Article 37, by computing the moment of half of the load-water-section relatively to that ordinate, and dividing by the area of that half section. To save time, the multiplication by one-third of the longitudinal interval may be omitted in computing both the moment and the area. A detailed example of those calculations will be given in Article 122.

[In symbols, let dx be an indefinitely small interval of the length of the load-water-section; x , its distance from ordinate No. 1; y , the corresponding ordinate of the load-water-line; and let the distance of the centre of flotation from ordinate No. 1, be denoted by \overline{AF} . Then,

$$\overline{AF} = \frac{\int xy dx}{\int y dx}.]$$

120. *Pitching and Scending of the Centre of Gravity.*—When the centre of flotation is not in the upright axis traversing the centres of gravity and of buoyancy, the motions of pitching and scending are accompanied by a rising and falling of the ship's centre of gravity, proportional, for a given angular motion, to the horizontal distance of the centre of flotation from the upright axis. For example, in Fig. 15, let AB represent the load-water-line of a



ship; G , her centre of gravity; C , her centre of buoyancy; ZZ , her upright axis, traversing those points. Let F be the centre of flotation, at the horizontal distance FH from the upright axis; then in pitching and scending, the vertical motions of all points in the vessel are performed as if she turned about a transverse axis traversing F ; so that with a given angular motion the centre of gravity undergoes a vertical motion to the following extent—

$$\overline{FH} \times \text{sine of angular motion};$$

and the vertical motion of the centre of gravity is $\begin{cases} \text{similar} \\ \text{contrary} \end{cases}$ to that of the vessel's bow, according as the centre of flotation lies $\begin{cases} \text{abaft} \\ \text{afore} \end{cases}$ the upright axis through the centre of buoyancy.

The centre of flotation lies almost always abaft the centre of buoyancy. This arises from the practice of making the ordinates of the after-body near the stern diminish more rapidly in going downwards from the load-water-line, than those of the fore-body; in other words, of making the "leanness" of the water-lines increase more rapidly in going downwards at the stern than at the bow.

As has been already stated in Article 112 and elsewhere, vertical motion of a ship's centre of gravity to any considerable extent is injurious both to the vessel and to her contents; and therefore, although in vessels for smooth-water navigation the position of the centre of flotation may be to a great extent arbitrary, it is necessary that in sea-going vessels that point should never be far distant from the upright axis.

The following are a few examples of the distance of the centre of flotation abaft the centre of buoyancy:—

	Feet.
Ships of the British Navy immediately before 1832, as given by Mr. Edey (two classes excepted, in which the distance was more than 3 feet),.....	1.1 to 2.7
Ships of the British Navy, classes introduced subsequently to 1832, Steamer, of which details will be given in Art. 122,.....	3.1 to 3.8
Two American clippers, described in Mr. Griffiths' Shipbuilder's Manual,.....	0.63
	0.23 and 0.02

121. The *Longitudinal Metacentre* is a point bearing the same relation to the pitching and scending motion of a ship which the metacentre for transverse motion, already described in Article 101, bears to the rolling motion. It is situated in the upright axis, ZZ , Fig. 12 (where it is represented by the point L); and its height, \overline{CL} , above the centre of buoyancy, C , is found by a process analogous to that of Art. 103; that is to say, by means of the Rules of Art. 43, the moment of inertia of the load-water-section relatively to the transverse axis through the centre of flotation, F , is computed; and that moment of inertia, being divided by the displacement, gives the required height, \overline{CL} .

The most convenient mode of proceeding is as follows:—Find, by Rule III. of Art. 43, the moment of inertia of half the water-section relatively to a transverse axis through A ; that is to say, multiply each ordinate by Simpson's multiplier, and by the square of the number of intervals by which it is distant from A ; add together the products, and multiply their sum by one-third of the cube of the longitudinal interval; the product will be the moment of inertia of half the water-section about A , which, multiplied by 2, will give that of the whole water-section.

Then, from the moment of inertia thus found, subtract the area of the water-section multiplied by the square of the distance of the centre of flotation from A (\overline{AF}^2); the remainder will be the required moment of inertia about F , which, being divided by the displacement in cubic feet, will give the height of the longitudinal metacentre above the centre of buoyancy.

[In symbols, using the same notation as in Art. 119,

$$\overline{CL} = \frac{\int x^2 y dx - \overline{AF}^2 \cdot \int y dx}{\int y dx dz}.$$

The symbol $\int y dx dz$ stands for half the displacement.]

In certain cases time may be saved by the following methods: if the centre of flotation happens to fall exactly upon an ordinate, that ordinate may be used instead of ordinate No. I. as the axis of moment of inertia; and if the water-lines are exactly similar at the bow and stern, the moment of inertia of one-quarter of the load-water-section may be computed relatively to the midship ordinate, and multiplied by 4.

To find the approximate *Longitudinal Righting Moment* for a given angle of pitch or scend, the centre of gravity of the ship, G, must be known by experiment or otherwise (see Art. 105), so that the height of the longitudinal metacentre above it may be found by subtraction, as follows:—

$$\overline{GL} = \overline{CL} - \overline{CG}.$$

Then the value of the moment in question is as follows:—

$$\text{Displacement} \times \overline{GL} \times \text{sine of angle.}$$

122. *Example.*—Calculation of the positions of the centre of flotation and the longitudinal metacentre of the load-water-section of the ship referred to in Articles 108, 109 and 117.

Number of Intervals.	Ordinates.	Simpson's Multipliers.	Products.	Multipliers for Leverage.	Products for Moments.	Products for Moment of Inertia.
1 ...	1 ...	1 ...	0.05 ...	0 ...	0 ...	0 ...
1½ ...	9 ...	2 ...	1.8 ...	½ ...	9 ...	4.5 ...
2 ...	1.9 ...	1 ...	1.9 ...	1 ...	1.9 ...	1.9 ...
2½ ...	2.9 ...	2 ...	5.8 ...	1½ ...	8.7 ...	13.05 ...
3 ...	4.0 ...	1½ ...	6.0 ...	2 ...	12.0 ...	24.0 ...
4 ...	6.2 ...	4 ...	24.8 ...	3 ...	74.4 ...	223.2 ...
5 ...	8.0 ...	2 ...	16.0 ...	4 ...	64.0 ...	256.0 ...
6 ...	9.3 ...	4 ...	37.2 ...	5 ...	186.0 ...	930.0 ...
7 ...	10.0 ...	2 ...	20.0 ...	6 ...	120.0 ...	720.0 ...
8 ...	10.0 ...	4 ...	40.0 ...	7 ...	280.0 ...	1960.0 ...
9 ...	9.7 ...	2 ...	19.4 ...	8 ...	155.2 ...	1241.6 ...
10 ...	8.7 ...	4 ...	34.8 ...	9 ...	313.2 ...	2818.8 ...
11 ...	6.9 ...	1½ ...	10.35 ...	10 ...	103.5 ...	1085.0 ...
11½ ...	5.5 ...	2 ...	11.0 ...	10½ ...	115.5 ...	1212.75 ...
12 ...	3.8 ...	1 ...	3.8 ...	11 ...	41.8 ...	459.8 ...
12½ ...	2.0 ...	2 ...	4.0 ...	11½ ...	46.0 ...	529.0 ...
13 ...	1 ...	½ ...	0.5 ...	12 ...	6 ...	7.2 ...
Sum 236.95				Sum 1523.7		11432.75
				Long. Int. 12.6		× 12.6
Divide by 236.95				19198.62		144,052.65
						× 12.6
Centre of Flotation abaft Section No. 1.....			AF = 81.02			1,815,063.39
Centre of Buoyancy abaft Section No. 1, } as in Articles 108 and 109.....			AB = 80.89			× 4.2
Centre of Flotation abaft Centre of Buoyancy.....			HF = 0.63			7,623,266.236
						× 2
Moment of Inertia relatively to ordinate No. 1.....						15,246,582.472
Deduct area of load-water-section, 1990.88 × (81.02) ²						13,065,332.807
Moment of Inertia relatively to ordinate through centre of } flotation.....						2,181,200.665

Which, being divided by the displacement, 7694 cubic feet, gives, for the height of the longitudinal metacentre above the centre of buoyancy, $\overline{CL} = 283.5$ feet.

123. By "trimming the vessel" is meant the operation of so adjusting the position of the lading lengthwise, that the bow and stern shall float with equal draught of water, or with any required difference of draught of water.

When the bow and stern are immersed to equal depths, the vessel is said to float "on an even keel;" and she is said to be trimmed "by the stern" or "by the head," according as the stern or the bow floats deepest. Vessels are almost always trimmed by the stern more or less.

The chief practical use of a knowledge of the position of the longitudinal metacentre, is to be able to calculate beforehand what alteration in the arrangement of a ship's lading will be necessary in order to produce any given change in her trim.

In Fig. 12, let AB be the original load-water-line, and A'B' the intended load-water-line after the alteration of trim, cutting the original line in F. The total required alteration of trim is the sum of the alterations of level at the bow and stern; that is, $\overline{AA'} + \overline{BB'}$. Let this be denoted by T. It will be divided between the bow and stern in the proportion of the distances of A and B from the centre of flotation, F.

From the longitudinal metacentre, L, draw LZ' perpendicular to the new water-line; this will be the new upright axis of the vessel, containing the new centre of buoyancy C', and the new centre of gravity G'. The distances through which those points are to be shifted may be found by calculation, as follows:—

$$\overline{CC'} = \frac{T \times \overline{LC}}{AB};$$

$$\overline{GG'} = \frac{T \times \overline{LG}}{AB}.$$

The distance $\overline{GG'}$ through which the centre of gravity is to be shifted, being multiplied by the displacement of the vessel, gives (according to a principle stated in Art. 61, and applied in Art. 105) the *moment necessary in order to produce the required alteration of trim*; that is to say, the product of the weight to be shifted into the distance through which its centre of gravity is to be shifted lengthwise; or if there be several weights to be shifted, the sum of the corresponding products for those weights.

Suppose, for example, that it is required to trim the vessel referred to in the last article *half a foot by the stern*, that is, to make the stern float half a foot deeper than the bow; and let it be assumed, that the centre of gravity is *two feet* above the centre of buoyancy. Then from previous calculations we have the following data:—

Length AB.....	= 151.2 feet
Distance of centre of flotation from bow.....	= 81.0 "
Height of longitudinal metacentre,	
Above centre of buoyancy, \overline{CL}	= 283.5 "
Above centre of gravity, \overline{GL}	= 281.5 "
Proposed alteration of trim, by the stern, T.....	= 0.5 foot
Displacement, 7694 cubic feet.....	= 219.8 tons.

From those data are computed the following results:—

$$\text{Elevation of the Bow, } \overline{AA'} = \frac{0.5 \times 81}{151.2} = 0.27 \text{ foot;}$$

$$\text{Depression of the Stern, } \overline{BB'} = \frac{0.5 \times 70.2}{151.2} = 0.23 \text{ foot;}$$

Centre of buoyancy to be shifted aft,

$$\overline{CC'} = \frac{0.5 \times 283.5}{151.2} = 0.94 \text{ foot;}$$

Centre of gravity to be shifted aft,

$$\overline{GG'} = \frac{0.5 \times 281.5}{151.2} = 0.93 \text{ foot;}$$

Required moment of shifted weight,

$$\text{Displacement} \times \overline{GG'} = 219.8 \times .93 = 204.4 \text{ foot-tons, nearly.}$$

123A.—Longitudinal Stability and Trim of very large Ships.—

If the length of the ship, AB (Fig. 15), be very great, and divided into a large number of intervals, it is convenient to take one of the ordinates situated near the centre of gravity of the load-water-section as the axis of moments, instead of the foremost ordinate, No. 1.

By this means large multipliers are avoided, and the calculations are separated into two parts.

This will appear from the following calculation of the longitudinal metacentre of H.M.S. *Warrior*, of which the drawings are given in this treatise.

The following particulars of the *Warrior* are required in calculating the alteration of her trim:—

Load Draught of Water,	Forward.....	Feet. Inches.
	Aft.....	25 0
		26 0

Displacement to the above draught = 301870 cubic feet = 8625 tons; length of the ship = 380 feet; interval between the ordinates, 18.3 feet.

The ordinate, No. 12, is situated near the centre of the ship, and it admits of Simpson's Rule being applied to the portions before and abaft it. In taking moments, therefore, the number of intervals between each ordinate and No. 12 ordinate must be taken as the multiplier, instead of the number of intervals from the ordinate No. 1.

LONGITUDINAL METACENTRE OF H.M.S. "WARRIOR."

FORE BODY.						
Number of Ordinates.	Ordinates.	Simpson's Multipliers.	Products.	Multipliers for Leverage.	Products for Moments.	Contin. Mults. for Mts. of Inertia.
1	1.2	$\frac{1}{3}$	0.6	11	6.6	11
1 $\frac{1}{2}$	4.1	2	8.2	10 $\frac{1}{2}$	86.1	10 $\frac{1}{2}$
2	7.2	1 $\frac{1}{3}$	10.8	10	108.0	10
3	13.8	4	55.2	9	496.8	9
4	19.4	2	38.8	8	310.4	8
5	23.5	4	94.0	7	658.0	7
6	26.1	2	52.2	6	313.2	6
7	27.7	4	110.8	5	554.0	5
8	28.6	2	57.2	4	228.8	4
9	28.9	4	115.6	3	346.8	3
10	29.0	2	58.0	2	116.0	2
11	29.0	4	116.0	1	116.0	1
12	29.0	1	29.0	0	0.	0
Totals.....			746.4		3340.7	20569.9
AFTER BODY.						
12	29.0	1	29.0	0	0.	0.
13	28.9	4	115.6	1	115.6	1
14	28.7	2	57.4	2	114.8	2
15	28.3	4	113.2	3	339.6	3
16	27.6	2	55.2	4	220.8	4
17	26.5	4	106.0	5	530.0	5
18	25.0	1 $\frac{1}{3}$	37.5	6	225.0	6
18 $\frac{1}{2}$	23.9	2	47.8	6 $\frac{1}{2}$	310.7	6 $\frac{1}{2}$
19	22.7	1	22.7	7	158.9	7
19 $\frac{1}{2}$	20.9	2	41.8	7 $\frac{1}{2}$	313.5	7 $\frac{1}{2}$
20	18.4	1	18.4	8	147.2	8
20 $\frac{1}{2}$	14.5	2	29.0	8 $\frac{1}{2}$	256.5	8 $\frac{1}{2}$
21	6.9	$\frac{1}{3}$	8.45	9	31.05	9
Totals,.....			677.05		2763.65	15367.6
Fore Body, do.,			746.4		3340.7	20569.9 Fore Body.
Long. Int.			Sum 1423.45	Diff. 577.05		35937.5
			3	6.1	L. Interval 18.3	18.3 Longitudinal Interval.
			8683.045	1423.45	10560.15	657656.25
						18.3 Longitudinal Interval.
						7.42 "
Cubic feet in a 35			(7)17366.09	Area of L.W. Sec.		12035109.375
Ton.			(5)2480.87	*Centre of Gravity of Load-Water Section before Ordinate No. 12.		6.1 Long. Interval \div 3.
			12)496.174			78414167.1875
Tons per Inch Immersion			= 41.3478			2
						146828334.375 Mt. of In. about Ord. 12.
						955134.95 Deduct Area \times (7.42) ² .
						301870)145873199.425 Mt. of In. about C. of G.
Area of L. W. Section =			17366.09			483.2 L. M. above C. of Buoy.
(7.42) ² =			55			8.2 C. of G. of Ship above do.
			955134.95			475.0 {Longitudinal Metacentre above Centre of Gravity.
Moment in foot tons to alter the trim one inch =						$\frac{475}{380} \times \frac{8625}{12} = 898.4$

EXAMPLE.—The *Warrior* is floating at her constructed draught of water as above given; and it becomes necessary to shift six guns on her main deck, each weighing 6 tons, 248 feet further aft. Required, the altered draught of water, forward and aft.

$$\text{Here } GG' (\text{Fig. 15}) = \frac{6 \times 6 \times 248}{8625} = 1.035 \text{ feet.}$$

Referring to the above table, it will be seen that the centre of gravity of the load-water-section is 7.42 feet before No. 12 ordinate, and is therefore $11 \times 18.3 - 7.42$ feet from the stem = 193.88. Also the distance of the same point from the stern =

$380 - 193.88 = 186.12$ feet. Now AA' , or the elevation of the bow, is equal to—

$$GG' \times \frac{AF}{GL} (\text{Fig. 15})$$

$$= 1.035 \times \frac{193.88}{475}$$

$$= .4224 \text{ foot} = 5 \text{ inches nearly.}$$

Also BB' , or the depression of the stern—

$$= 1.035 \times \frac{186.12}{475}$$

$$= .4055 = 4\frac{7}{8} \text{ inches nearly.}$$

The new draught of water will therefore be—

	Feet.	Inches.
Forward.....	24	7
Aft.....	26	4 $\frac{7}{8}$

In ships of the usual form, a water-section parallel to the load-water-section situated either a few inches above or below that section, will not differ much from that section in area, in the position of its centre of gravity, or in its moment of inertia about a similar axis.

Consequently, from the calculations above made, the seat of a ship in the water may be found accurately enough for all practical purposes, either when a given weight is placed on board of a ship in a given position, or when a given weight is taken out of a ship from a given position on board.

1st, When a given weight is to be placed on board of a ship in any position—

Let P be the weight in tons, and b the horizontal distance of its centre of gravity, in a fore-and-aft direction from the centre of gravity of the load-water-section.

If the weight, P , be placed on board the ship directly over the centre of gravity of the load-water-section, it is clear that the ship will sink down in the water until the weight of the additional water displaced is equal to P , and the new water-line will be parallel to the original water-line.

Then, if the weight, P , be shifted to the position intended, distant b feet from the centre of gravity of the load-water-section, the problem resolves itself into the question of the alteration of the trim when a given weight on board is moved in a fore-and-aft direction, setting out from the water-line, at the deep draught, parallel to the original water-line.

EXAMPLE.—A weight of 70 tons has to be placed on board of the *Warrior*, when floating at her constructed draught of water. The centre of gravity of the additional weight is to be situated at a distance of 80 feet before the centre of gravity of the load-water-section. Required the new draught of water, forward and aft.

Referring to the table already given, it is found that the displacement per inch immersion is 41.3478 tons. The increased immersion due to the 70 tons added will therefore be—

$$\frac{70}{41.3478} \text{ inches} = 1.693 \text{ inches;}$$

and the new draught of water will be—

	Feet.	Inches.
Forward.....	25	1.693
Aft.....	26	1.693

Also, since the additional displacement has its centre of gravity in the middle between the original and the new water-lines, the new centre of buoyancy will be found to be .06 foot above the original centre of buoyancy; also, assuming the centre of gravity of the weight added to be about 18 inches above the original water-line, the new centre of gravity of the ship will be .01 foot above the original centre of gravity. The new centre of buoyancy

will, therefore, be .06 — .01, or .05 nearer to the new centre of gravity of the ship than the original centre of buoyancy was to the original centre of gravity. The distance between the centres of gravity and of buoyancy originally was 8.2 feet (see the above table); consequently, the distance between the new centres of buoyancy and gravity is 8.15 feet.

Again, since the moment of inertia of the new plane of flotation is practically the same as that of the original plane of flotation, the height of the new longitudinal metacentre above the new centre of buoyancy will be to the height corresponding to the original water-line as the original displacement is to the new displacement; or it is equal to—

$$483.2 \times \frac{8625}{8695} = 479.3.$$

Consequently the height of the new longitudinal metacentre above the new centre of gravity
 $= 479.3 - 8.15 = 471.15.$

When the weight (70 tons) is moved forward to a distance 80 feet before the centre of gravity of the load-water-section, the centre of gravity will move forward through a distance, GG' (Fig. 15), equal to $\frac{70 \times 80}{8695}$ feet.

Assuming that the depression of the bow will be practically equal to the elevation of the stern—

The depression of bow = elevation of stern

$$= GG' \times \frac{\text{Half the length of the Ship}}{\text{Height of the Longitudinal Metacentre}}$$

$$= \frac{70 \times 80}{8695} \times \frac{190}{471.15} \text{ in the example.}$$

Putting for GG' the value given above—

$$\text{Depression of the bow} = \frac{1064000}{4096649.25} \text{ foot} = .26 \text{ foot} = 3.12 \text{ inches.}$$

The new draught of water of the *Warrior* will therefore be—

	Feet.	Inches.	Inches.	Feet.	Inches.
Forward,.....	25	1.693	+ 3.12	= 25	4.813
Aft,.....	26	1.693	— 3.12	= 25	10.573

The alteration of the trim of a ship caused, by the removal of some of her weights may also be found in exactly the same manner.

The calculations for finding the alteration of the trim of a ship may, however, be much abridged from the following considerations:

—It has already been proved, when treating of transverse stability (Article 117A), that if a ship floating at a given draught of water has either additional weights put on board, or some of her weights taken out of her, the centre of gravity of the weights put on board or taken out being situated in the same horizontal plane with the centre of gravity of the displacement added or subtracted; and if, moreover, the form of the water-section in every case is the same (that is, if the sections of the ship “between wind and water” be vertical)—then the transverse stability in the three cases is precisely the same. In the same manner it may be proved, that in a ship of the same form, if additional weights be put on board, or some of her weights taken out, and the centre of gravity of the weights added or subtracted be situated in the same horizontal

plane with the centre of gravity of the displacement added or subtracted, then the moment required to alter the trim to any given amount will remain unaltered.

The centre of gravity being usually situated in the neighbourhood of the load-water-line, and its position only approximately known, the results will be sufficiently near for practical purposes, if we disregard the very small variation in the position of the centre of gravity of the ship caused by the centre of gravity of the weights, added or subtracted, not being situated in the same horizontal plane with the centre of gravity of the displacement added or subtracted. It will then only be necessary to calculate the moment required to alter the trim of a ship of the usual form (say one inch) when floating at her load draught of water, and the same may then be taken for any other draught of water not differing very much from the load draught.

When the trim is altered one foot—

$$GG' \text{ (Fig. 15)} = 1 \times \frac{GL}{\text{Length of Ship (A B)}};$$

And the moment to alter the trim one foot, which is

$$= \text{Displacement} \times GG'$$

$$= \text{Displacement} \times \frac{GL}{AB}$$

$$= W \times \frac{GL}{AB} \text{ (Putting W for displacement);}$$

And the moment to alter the trim one inch

$$= \frac{W}{12} \times \frac{GL}{AB};$$

And the moment to alter the trim n inches

$$= n \times \frac{W}{12} \times \frac{GL}{AB}.$$

Hence if the moment of the weights moved be given, and the moment required to alter the trim one inch has been calculated, the alteration of the trim in inches

$$= \frac{\text{Given moment of weights moved}}{\text{Moment to alter the trim one inch}}.$$

Taking the case of the *Warrior* at her load draught of water; see foregoing table (page 60)—

Moment to alter the trim one inch

$$= \frac{8625}{12} \times \frac{475}{380} = \frac{4096875}{4560} = 898.4.$$

The alteration of the trim, or $AA' + BB'$, in Fig. 15, caused by the removal of 70 tons through a horizontal distance of 80 feet

$$= \frac{70 \times 80}{898.4} \text{ inches}$$

$$= 6.2 \text{ inches;}$$

and since the depression at the bow is nearly equal to the elevation of the stern, each is = 3.1 inches; and supposing that the weights be removed aft, the new draught of water will be—

	Feet.	Inches.
Forward,.....	24	9
Aft,.....	26	3

very nearly.*

* For details as to the subject of this Article, see a paper by Mr. Barnes “On the Longitudinal Metacentre; its use in calculating Trim; and a New Method of finding it,” read to the Institution of Naval Architects in March, 1864.

CHAPTER IV.

OSCILLATIONS OF SHIPS.

124. *Subjects and General Principles of this Chapter stated.*—The oscillations of a ship may be divided into two main classes, distinguished as *free* and *forced*.

Free oscillations are those movements to and fro about her steady position, which a ship performs in smooth water, when displaced by the temporary action of some disturbing force, and then left to herself; and their extent and period depend on the disturbing force, and on the dimensions, figure, and distribution of the weight of the ship, according to the general laws of oscillatory movement already stated in Chap. II., Art. 83.

Forced oscillations are those in which the ship is compelled to accompany the water in which she floats, while its particles move so as to form waves. The movements of the particles of water also depend on the principles of Article 83, together with those of Articles 81 and 84.

The actual oscillatory movements of a ship are always compounded of free and forced oscillations, in a more or less complicated manner; and it often happens that the very qualities which tend to prevent oscillations of one kind, tend at the same time to augment those of another kind; so that it is a difficult problem to design a ship in which neither the free nor the forced oscillations shall be too extensive or too frequent. This fact has already been referred to in Chapter I., Articles 5 and 6.

The reaction of the particles of a ship while she performs oscillatory movements, taking place according to the laws stated in Article 84, produces various straining effects upon her structure, whose nature and amount, and the means of resisting them, will be discussed in the Third Division of this treatise. That straining tendency of oscillatory movements of all kinds is one of the principal reasons for studying how to keep them within moderate limits as to frequency and extent.

In treating of any particular sort of oscillation of ships and waves, it is often convenient to refer to the height of the "*equivalent revolving pendulum*" whose time of revolution is the same with the period of the oscillation. The laws of the motion of revolving pendulums have been stated in Chap. II., Art. 82.

It is to be observed that by the *period* of an oscillation is always meant, the time occupied by a complete or double oscillation, which brings the oscillating body back to the condition from which it started, both as to position and as to direction of motion.

In Chap. II., Article 83, it has already been stated that in order that the oscillations of a body may be *isochronous*—that is, performed in equal periods, whatsoever their extent may be, it is necessary that the righting force or righting moment should be simply proportional to the extent of the disturbance of the body from the steady position. That condition is scarcely ever fulfilled exactly by ships; but in general it is so nearly fulfilled, that no error in any practical conclusion is caused by treating the oscillations of ships as if they were isochronous; and they will accord-

ingly be so treated in what follows; except that, in a note, it will be shown how the periodic time of rolling of a ship which does not roll isochronously may be computed if required.

The present Chapter is divided into three sections. The first treats of the Free Oscillations of Ships; the second, of the main facts and principles of the Motion of Waves; and the third, of the Forced and Compound Oscillations of Ships.

SECTION I.—FREE OSCILLATIONS OF SHIPS.

125. The *Different Kinds of Free Oscillation* of which a ship is capable are three in number, viz.—*Rolling*, or oscillation from side to side about a longitudinal axis:—*Pitching*, or oscillation in a vertical plane about a transverse axis:—and *Dipping*, which term has been introduced to denote a vertical oscillation of the vessel as a whole, consisting in alternate rising above and sinking below the position of steady flotation. Those three kinds of oscillation may be combined in various ways.

126. *Rolling.*—When rolling in smooth water is not accompanied by pitching or dipping, it is performed about a longitudinal axis traversing the ship's centre of gravity. When the periodic time of rolling is not affected by the resistance opposed by the water to motion of this kind, it depends on the principles of Article 83 in the following manner:—

According to Rule II. of that Article, we have—

$$\begin{aligned} & \text{Height of the Equivalent Pendulum} \\ = & \text{Moment of Inertia of the weight of the ship} \times \text{Angle of Heel in} \\ & \text{circular measure} \\ & + \text{Moment of the Righting Couple.} \end{aligned}$$

But the moment of inertia of the ship's weight about a longitudinal axis is equal to her displacement multiplied by the square of her transverse radius of gyration (Art. 78); and the moment of the righting couple is *nearly* equal to the displacement, multiplied by the height of the metacentre above the centre of gravity, multiplied by the angle of heel in circular measure (Article 104). Dividing, therefore, both the dividend and the divisor of the preceding expression by the displacement, and by the angle of heel, the following rule is obtained:—

I. *Divide the square of the ship's transverse radius of gyration by the height of her metacentre above her centre of gravity; the quotient will be the Height of her Equivalent Pendulum for Rolling.*

Then from Chap. II., Article 82, we have the following rule:—

II. *Divide the height of the equivalent pendulum in feet by .815; the square root of the quotient will be the period of a double roll in seconds.*

[In algebraical symbols, let m be the height of the metacentre above the centre of gravity in feet; r , the transverse radius of gyration in feet; T , the periodic time of rolling in seconds: then—

$$T = \sqrt{\frac{r^2}{.815 m}}]$$

The following is the geometrical construction for finding the equivalent pendulum. In Fig. 1, let G represent the ship's centre of gravity (as to the determination of which, see Article 105), and M her metacentre. Perpendicular to GM, draw \overline{GR} , equal to the transverse radius of gyration. Join MR; and perpendicular to it draw RP, cutting MG produced in P; \overline{GP} will be the height of the equivalent pendulum required.

To represent a compound pendulum which shall have not only the same period of oscillation with the ship, but the same statical and dynamical stability, proceed as follows. About M, with a radius equal to GR, draw a circular arc cutting the straight line, RG, in two points, B, B, equidistant from G. Conceive that MBB represents a light triangular frame, hung from the point M; and that it is loaded by having one half of the weight of the ship concentrated at each of the points, B, B: the triangular frame so suspended and loaded will be the compound pendulum required.

It is so difficult to determine, by mere measurement and calculation, the radius of gyration of so complex a combination of weights as a ship, that the two preceding rules can seldom be used directly. The converse rules, for deducing a ship's radius of gyration from experiments on her period of rolling, are more easily applied. Their use will afterwards appear. They are as follows:—

III. Let the ship, when floating in smooth water, be forcibly heeled over through a moderate angle, and left to roll freely; count the number of double rolls in a convenient interval of time; divide the time by the number, for the period of rolling. Then, the square of the period in seconds, multiplied by .815, will be the height of the equivalent pendulum in feet.

IV. Multiply the equivalent pendulum by the height of the ship's metacentre above her centre of gravity; the product will be the square of her radius of gyration.

[In algebraical symbols;—

$$r^2 = .815 m T^2.]$$

For example, suppose the following experiment to have been made:—A ship, having been heeled over in smooth water, and set free to roll, made 15 double rolls in 120 seconds; so that her period of rolling was—

$$\frac{120}{15} = 8 \text{ seconds.}$$

Then the height of the equivalent pendulum was—

$$8^2 \times .815 = 52.16 \text{ feet;}$$

and if the height of her metacentre above her centre of gravity was 4 feet, the square of her transverse radius of gyration was—

$$4 \times 52.16 = 208.64;$$

and her transverse radius of gyration itself was—

$$\sqrt{208.64} = 14.44 \text{ feet.}$$

126A. *Isochronous-Rolling Vessels.*—It has already been explained in the introductory Article of this Chapter (Art. 124), that the oscillations of ships will be treated as sensibly isochronous, or performed in uniform periodic times, whatsoever the extent of the oscillation; such being very nearly the case in practice. It is desirable, however, to show what the form of a ship ought to be, in order that her rolling may be *exactly* isochronous.

In order that any oscillation may be isochronous, the righting moment should be proportional simply to the angle of disturbance. Hence, in the case of a ship, the arm of the righting couple, which is the perpendicular distance from the centre of gravity to a vertical line through the centre of buoyancy, should be simply proportional to the angle of heel.

In a paper communicated to the Institution of Naval Architects, in 1864, the following propositions are demonstrated, for an isochronous-rolling ship:—

The metacentric evolute (see Chap. III., Art. 118) is the involute of a circle described about the centre of gravity and through the metacentre; and consequently, the metacentric involute is the involute of the involute of that circle.

[In the same paper are also demonstrated the following propositions as to isochronous rolling ships, which cannot well be expressed without the aid of algebraical symbols.

Let m denote the height of the metacentre above the centre of gravity;

b the height of the metacentre above the centre of buoyancy, in the upright position;

b' , the radius of curvature of the metacentric involute, when the angle of heel is θ (in circular measure);

y , the half breadth of the upright water-section;

y' , the half breadth of the inclined water-section, for the angle of heel θ .

Then

$$b' = b + \frac{m \theta^2}{2}; \dots \dots \dots (1)$$

$$y' = y \left(1 + \frac{m \theta^2}{2b}\right)^{\frac{1}{2}}. \dots \dots \dots (2).$$

To approximate to the form of cross section between wind and water, by a pair of circular arcs, let—

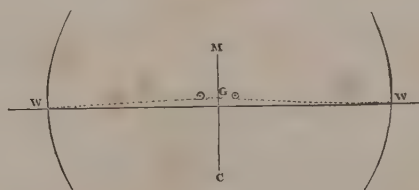
a denote the perpendicular distance of the centre of gravity from the upright plane of flotation; then the proper radius for the pair of arcs will be—

$$\sqrt{y^2 + a^2} \cdot \left(1 + \frac{m}{3b}\right). \dots \dots \dots (3).$$

At the point where each arc cuts the upright water-line, its radius should be a line passing through the centre of gravity.]

The construction above mentioned is shown in Fig. 1A, in which G represents the ship's centre of gravity; M, her meta-

Fig. 1A.



centre; C, her centre of buoyancy; WW, the upright water-line. Draw the two straight lines, WG, WG; produce them beyond G; and take in each of them,—

$$\overline{GO} = \overline{WG} \cdot \frac{\overline{GM}}{3 \overline{CM}};$$

then the two points marked O, O, will be the centres for two circular arcs through the points W, W, each centre belonging to the arc furthest from it; and if those arcs be taken for the cross

sections of a ship between wind and water, she will roll isochronously.*

127. Regulation of Period of Rolling.—A ship's period of rolling is to a certain extent capable of regulation by suitable stowage of the weights which she carries. For reasons which will be explained in the Third Section of this Chapter, the object of such regulation is usually to *lengthen* that period as far as may be consistent with sufficient statical stability.

There are two means of lengthening the period of rolling: to raise the centre of gravity of the ship nearer to the metacentre; and to increase her radius of gyration. But when the centre of gravity has been brought as near to the metacentre as is consistent with sufficient statical stability (that is, about four feet below the metacentre, as stated in Article 105), the period of rolling cannot be lengthened further by that means, and there remains only the lengthening of the transverse radius of gyration.

The lengthening of the transverse radius of gyration without altering the position of the ship's centre of gravity is effected by what is called "*winging out*" the weights carried by the ship: that is, spreading them sideways.

The effect of winging out given weights to a given extent is thus estimated agreeably to the principles of Art. 78. Suppose a pair of equal weights to be originally placed with their centres of gravity at given equal distances at opposite sides of the longitudinal middle plane of the vessel, and that they are shifted horizontally outwards to greater but still equal distances from that plane. Then—*From*

* The Exact Period of Unresisted Rolling of a ship which is not isochronous may be computed, if required, by a method first investigated by the Reverend Canon Moseley.** The general principle of that method cannot well be expressed without the aid of algebraical symbols; but rules for calculation will be added in words. In what follows, the ship is supposed to roll without vertical motion of her centre of gravity.

[Let θ . denote the extreme angle of heel;

θ . any less angle;

W , the weight of the vessel;

r , her transverse radius of gyration;

z_1 , the depression of her centre of buoyancy relatively to her centre of gravity considered as fixed, at the extreme angle of heel θ_1 ;

z , the corresponding depression, for the less angle θ ; (so that, in short, Wz_1 and Wz are respectively the values of the dynamical stability of the vessel at the angles θ_1 and θ);

t , time in seconds, reckoned, while the ship is righting herself, from the instant when she reached the extreme angle θ_1 ;

then by the time the vessel has partially righted herself by heeling from θ_1 to θ , the potential energy exerted on her by the water is—

$$W(z_1 - z);$$

and the actual energy acquired by the vessel is —

$$\frac{W r^2}{2g} \cdot \frac{d\theta^2}{dt^2};$$

and those quantities must be equal; therefore—

$$\frac{d\theta^2}{dt^2} = \frac{2g(z_1 - z)}{r^2}; \text{ and } \frac{d\theta}{dt} = \sqrt{\frac{2g(z_1 - z)}{r^2}};$$

This being integrated between the limits $\theta = \theta_1$, and $\theta = 0$, gives the time occupied by the vessel in righting herself, which time is one-quarter of the periodic time of a double roll; therefore, let T denote that periodic time; then—

$$T = \frac{4\pi}{\sqrt{2g}} \cdot \int_0^{\theta_1} \frac{d\theta}{\sqrt{z_1 - z}}$$

The value of the coefficient $\frac{4\pi}{\sqrt{2g}}$ is very nearly $\frac{1}{2.006}$, or 0.4985.]

The preceding formula, expressed in words, is as follows:—

I. Divide the extreme angle of heel into a convenient number of equal intervals, expressed in circular measure (Art. 30).

II. Find, according to the principles of Art. 116, the depression of the centre of buoyancy relatively to the centre of gravity considered as fixed, which is produced by the extreme angle of heel, and by each of the intermediate angles of heel respectively.

III. Subtract each of the smaller depressions from the greatest depression, and take the reciprocals of the differences, and the square roots of these reciprocals.

IV. Consider those square roots as the ordinates of a curve, whose base represents the extreme angle of heel; and find the area of that curve by one of Simpson's rules.

V. Multiply that area by the ship's transverse radius of gyration, and divide (for British measures) by 2.006. The result will be the periodic time of a double roll in seconds.

** Philosophical Transactions, 1850:—also Mechanics of Engineering and Architecture, Second Edition, Appendix, Note C.

the square of the new distance subtract the square of the original distance; multiply the remainder by the sum of the shifted weights, and divide by the displacement; the quotient will be the increase of the square of the ship's transverse radius of gyration.

The period of rolling will be increased in the simple ratio of the radius of gyration. For example, suppose that a ship of 800 tons displacement has engines and boilers weighing 200 tons, which are divided into two equal halves by the longitudinal middle plane of the ship; and that those halves are found to have their centres of gravity 5 feet from that plane at opposite sides. Suppose that by a new arrangement of engines and boilers, of the same total weight, the centres of gravity of those two halves can be winged out 5 feet each way, so as to increase their distances from the middle plane to 10 feet. Then the increase in the square of the transverse radius of gyration will be—

$$(10^2 - 5^2) \times \frac{200}{800} = 18.75.$$

If the original radius of gyration of this vessel was 10 feet, her new radius of gyration will be—

$$\sqrt{100 + 18.75} = 10.9 \text{ feet};$$

and her period of rolling will be lengthened in the ratio of 10.9 to 10.

A ship's transverse radius of gyration may be shortened if required, so as to shorten her period of rolling, by concentrating the weights amidships, being the reverse of the operation described above.

A ship's period of rolling is shortened by the lowering of her centre of gravity relatively to her metacentre; for example, by placing heavy lading or ballast in the hold. To lessen or prevent, if necessary, this effect of a heavy cargo, is one of the uses of *dunnage*, already mentioned in Chap. III., Art. 113.

The loss of her masts at once lowers a ship's centre of gravity, and shortens her radius of gyration; and hence a dismasted ship rolls quickly.

There is one position in which additional weights can be put on board a ship without altering the depth of her centre of gravity below her metacentre; and that is, when the centre of gravity of such additional weights is situated exactly at the same depth below the centre of the additional volume of water displaced, that the original centre of gravity of the ship is below her original metacentre; for then the centre of gravity and the metacentre are equally lowered. (See Article 117A).

128. Rolling as affected by the Passive Resistance of the Water.—In the numerical rules given in the preceding Article, the effects of the passive resistance of the water on the periodic time of rolling of the vessel, are not taken into account.

By the *passive resistance* of the water is to be understood that force which does not tend to right the vessel, or turn her back to a definite position, but merely opposes her motion in what direction or manner soever she may be moving. One effect of passive resistance on free oscillations of all kinds is gradually to diminish their extent, and finally to bring them to an end; and but for this, they would go on undiminished for ever, after having once been set going.

The resistance of a fluid to the motion of a body immersed in it consists principally of two parts: one of those parts is the direct effect of the resistance of the fluid particles to sliding past each other, and is proportional to the velocity of the motion of the immersed body simply; the other part arises from the

waste of mechanical work in producing eddies in the fluid, and is proportional to the square of that velocity.

At low velocities, the first part of the resistance (varying as the velocity simply) is the greater part, and at very low velocities the only part appreciable. It lengthens the period of rolling in a constant proportion, and makes the extent of rolling diminish in geometrical progression.*

At high velocities, the second part of the resistance (varying as the velocity squared) is the greater part; and at very high velocities the first part becomes insignificant in comparison with it. It does not affect the period of rolling; but it causes the extent to diminish in harmonic progression; that is to say, in proportion to some set of equidistant terms of the series of reciprocals, $1, \frac{1}{2}, \frac{1}{3}, \frac{1}{4}, \frac{1}{5}, \&c.$ †

It is obvious that what may be called the *steadying action* of the passive resistance of the water, in diminishing the extent and lengthening the periodic time of rolling, must be comparatively small in ships with flat floors, rounded cross sections, and little or no keel, and comparatively great in ships with rising floors and deep keels; and those general facts, indeed, are well known through practical experience; but there is a want of precise data as to the amount of that action on vessels of particular forms, dimensions, and proportions.

In the paper of 1864 referred to below, rules for conducting some such experiments, in case they should ever be made are given in the following terms:—

“By means of suitable experiments on the rolling of a ship in smooth water, two quantities may be determined—the *square of her radius of gyration*, and a quantity which may be called the *leverage of keel-resistance*; being the length of the lever at which the whole weight of the ship would have to act, in order to exert a moment equal to the moment of the resistance opposed by the water to the keel, when the angular velocity of rolling is unity.

“I. Let the ship be forcibly heeled over, and set free to roll; observe the periodic time of rolling by counting the complete oscillations or *double rolls* in a certain number of seconds; observe also the greatest angle of heel at the commencement of the experiment, and also after the lapse of a certain time in seconds: taking care to measure those angles by observations of fixed objects, or by an instrument of the gyroscope kind (like that invented by Professor Piazzzi Smyth), and not by a plummet or level.

“II. Divide the hyperbolic logarithm of the ratio in which the original angle of heel exceeds the diminished angle, by the time in seconds (or the common logarithm by the time in seconds $\times .4343$); the quotient will be a number which we may call the *exponent*.

“III. To find what the periodic time would be in the absence of keel-resistance:—Multiply the square of the actual periodic time of a double roll in seconds and fractions of a second, by the square of the *exponent* above mentioned, and divide the product by $39.48 (4\pi^2)$; to the quotient add 1; then by that sum divide

the square of the actual periodic time; the result will be the *square of the periodic time of unresisted rolling*.

“IV. Multiply the square of the periodic time of unresisted rolling by the constant $0.815 (\frac{g}{4\pi^2}$ in feet); the product will be the length of the *corresponding simple pendulum* in feet.

“V. Multiply that length by the height of the ship's metacentre above her centre of gravity; the product will be the *square of her radius of gyration*.

“VI. Multiply the square of the radius of gyration by the *exponent* (Rule II.), and divide by $16.1 (\frac{g}{2}$ in feet); the quotient will be the *leverage of keel-resistance* in feet.

“Experiments and calculations of the kind just described are most likely to give accurate and consistent results at moderate angles of heel (say, not exceeding about 10°); for it is only under that condition that the resistance to rolling can be treated as approximately proportional to the angular velocity of rolling. The test whether the angles of heel are small enough is simply their diminishing sensibly in geometrical progression.”

In long vessels with fine ends, the sharpness of the entrance and run are to a certain extent relied on to make up in steadying action for roundness and flatness amidships.

In order to combine the convenience of a flat floor with sufficient resistance to rolling, “bilge-keels” are sometimes used; but their efficiency is doubtful.‡

The subject of the use of keels, bilge-keels, and lee-boards in resisting lee-way, belongs to Chapter V. of this Division.

129. *Rolling as affected by the Wind—Lurching*.—Although the wind, by raising waves in the sea, is the indirect cause of most of the oscillations of ships, and although its action in sudden gusts produces rolling, the direct effect of its steady action on a ship's sails is on the whole to promote steadiness, and to keep the ship heeled over at a nearly constant angle, depending on her statical stability, and on the moment of the transverse component of the pressure of the wind on the sails (which will be further considered in Chapter V.)

The wind also promotes steadiness in the vessel indirectly, by producing *lee-way*: which consists in a drifting of the ship through the water in a direction transverse to the keel, and has the effect of increasing the resistance opposed by the water to the ship's rolling.

Both the direct and the indirect steadying actions of the wind are greater when the ship's course lies near the wind; and they both disappear when she runs before it.

The rolling oscillations of a ship which is heeled over by the action of the wind, instead of being of equal extent to either side of a vertical plane, are of equal or nearly equal extent to either side of a plane inclining to leeward of the vertical at the angle of heel corresponding to the steady pressure of the wind. Under these circumstances, when the ship rolls to leeward she is said to *lurch*; and when she rolls to windward, to *heel*; the

* The effects of this first part of the resistance of fluids on the motions of immersed bodies in general, and especially on the oscillations of pendulums, and on the motion of waves, are thoroughly investigated in a paper by Professor Stokes, in the *Cambridge Transactions* for 1850. The theory of its effects on the rolling of ships is given in a paper by the Editor of this Treatise, communicated to the Institution of Naval Architects in 1864.

† The effects of the second part of the resistance of fluids on the rolling of ships are explained in a paper by Mr. Froude, in the *Transactions of the Institution of Naval Architects* for 1862.

‡ Between 1770 and 1780, an attempt was made to combine in a ship the convenience of a flat floor with the steadying action of a sharp floor, by means of a peculiar form of vertical section, like that shown in Fig. 1a. A favourable account of her sailing and steadiness is given in the papers of the Society for the Improvement of Naval Architecture, published in 1792; but it is not altogether satisfactory, being given at second-hand, and deficient in detail.

Fig. 1a.



word "heel" having this restricted meaning when used in contradistinction to "lurch."

The periodic time of lurching and heeling to either side of an inclined plane is, in well-formed ships, *nearly* equal to the periodic time of rolling to either side of a vertical plane; it is not, however, *exactly* equal, unless the ship is so formed as to roll isochronously, according to the principle stated in Article 126A. Should the rolling of the ship not be isochronous, the safest side on which to deviate from isochronism is, that the periodic time should be shorter when the middle plane of the oscillations is inclined than when it is vertical: consequently it is safer to make the radius of curvature of a cross section between wind and water greater than that given by the rule of the Article referred to, than to make it less.

The utmost probable inclination to which a well-formed ship may be carried by lurching, with wind not exceeding a certain strength, is about double the angle corresponding to the steady pressure of the same wind; such being the effect of a total lull of the wind, followed by a sudden rising to its original strength.

130. *Pitching* is an oscillation performed by the ship in a vertical plane, about a transverse axis. The single word "pitching" is commonly used to denote the complete or double oscillation; although when the two parts of which it consists are spoken of separately, the forward part of the oscillation, or that which lowers the bow and raises the stern, is called "pitching;" and the backward part, or that which raises the bow and lowers the stern, "scending," as already mentioned in Arts. 114 and 120.

It is unnecessary to state in detail the principles which regulate the periodic time of pitching; for they are precisely similar to those which have already been stated in Art. 126, with reference to rolling oscillations; and the four rules given in that Article are all made applicable to pitching oscillations, by the following modifications:—

For the height of the transverse or ordinary metacentre above the centre of gravity, substitute the height of the *longitudinal metacentre* (as to the finding of which, see Articles 121, 123A).

For the square of the transverse radius of gyration, substitute *the square of the longitudinal radius of gyration*; that is to say, a quantity found by multiplying each weight in the ship by the square of its distance from a transverse axis through the centre of gravity, adding together the products, and dividing by the total weight, according to the principles of Article 78.

A very thin flat raft, consisting wholly of some buoyant material of uniform heaviness, has its periodic times of pitching and rolling exactly equal; for the height of the metacentre for a given axis of oscillation is equal to the geometrical moment of inertia of the water-section about that axis divided by the volume of displacement; and the square of the radius of gyration is equal to the mechanical moment of inertia of the raft about the same axis, divided by the weight of the displacement; but the mechanical moment of inertia, for a thin flat raft, is equal to the geometrical moment of inertia multiplied by the depth of immersion and the heaviness of water; and the weight of the displacement is equal to its volume multiplied by the heaviness of water; whence it follows, that the square of the radius of gyration of a thin flat raft about any given axis is equal simply to the height of the metacentre for that axis, multiplied by the depth of immersion; and consequently, that *the equivalent pendulum of a thin flat raft, for all horizontal axes of oscillation, is equal to the depth*

of immersion; and that its periodic times of pitching and rolling are equal.

The distribution of weight and that of volume in almost all ships differ from those in a raft in such a way, that the periodic time of pitching is in general much shorter than that of rolling.

In vessels of small depth and shallow draught, with light masts or none, the periodic time of pitching is nearly that of a pendulum whose height is the mean depth of immersion; but that of rolling is considerably longer.

131. *Regulation of Period of Pitching—Liveliness.*—For reasons which will appear in the third Section of this Chapter, it is in most cases desirable to shorten the periodic time of pitching as far as practicable, so that the vessel may be what is called *lively* in pitching. There are two means of doing so—to increase the height of the longitudinal metacentre, and to shorten the longitudinal radius of gyration. The height of the longitudinal metacentre is usually so great that no practicable alteration of the level of the centre of gravity can have any material effect upon it; the only way, therefore, in which it can be increased in the construction of the vessel, is by giving fullness of form to her ends between wind and water. But there are limits to the fullness of a ship's ends which cannot be passed without detriment to speed and economy of power. There remains then, as a means of insuring liveliness in pitching, the shortening of the longitudinal radius of gyration, which is effected by concentrating the weights on board lengthwise towards the centre of gravity, as closely as may be practicable or convenient.

132. *Dipping* is a name which may be applied to an alternate rising and sinking of the whole ship, out of and into the water. Applying to this kind of oscillation the principle of Article 83, we find that the righting force is the weight of water contained in a layer whose area is the plane of flotation, or load-water-section, and its thickness the *dip* itself, or vertical disturbance; whence we have the proportion—

As the layer disturbed,

: is to the whole displacement,

: : so is the extent of dip

: to the *mean depth of immersion* (Chap. III., Art. 98), which is therefore the *equivalent pendulum* for dipping oscillations.

Hence the following rules:—

I. *Divide the volume of displacement by the area of the load-water-section; the quotient will be the height of the equivalent pendulum for dipping oscillations.*

II. *Divide that height, in feet, by .815; the square root of the quotient will be the periodic time of a complete dipping oscillation, consisting of a dip and rise.*

For example, take the following data:—

Load Displacement,.....360 tons = 12600 cubic feet;
Area of Load-water-section,..... 2100 square feet;

whence we have—

$$\frac{12600}{2100} = 6 \text{ feet, height of equivalent pendulum for dipping;}$$

$$\sqrt{\frac{6}{.815}} = \sqrt{7.37} = 2.71 \text{ seconds, periodic time of dipping oscillations.}$$

From what has been stated in Article 130, it appears that for a thin flat raft, the periodic times of all the three kinds of free oscillation, rolling, pitching, and dipping, are the same. But in actual vessels the period of dipping is shorter than that of pitching, and much shorter than that of rolling; and for that reason, as well as for other reasons which will be explained in the Third Division, it

strains the ship and damages her contents more than any other kind of oscillation. Ships which are subject to it are said to be *uneasy*.

Dipping oscillations are seldom produced directly by an external force, but are the indirect effect of other kinds of oscillation, through unskilful construction or stowage of the ship. They belong to the class of *secondary oscillations*, which will be considered in the next Article.

133. *Secondary Oscillations.—Easy and Uneasy Motion.*—"Secondary oscillations" is a name which may be applied to oscillations which are the effect of other oscillations, owing to some fault in the figure of the ship or the distribution of her weights. They are of three principal kinds—Pitching produced by Rolling, Dipping produced by Rolling, and Dipping produced by Pitching. In every case in which the period of a secondary oscillation is either the same, or commensurable with, that of the primary oscillation by which it is produced, and there is a long series of such primary oscillations, the extent of the secondary oscillation may go on increasing for a considerable time.

I. *Pitching produced by Rolling, or Secondary Pitching*, arises from the centres of the wedges of immersion and emersion not being in the same transverse plane. Its cause, and the means of preventing it, have been fully explained in Chap. III., Article 114. According to Fincham ("Outline of Shipbuilding," Part I.), the longitudinal distance between the centres of the wedges, in easy ships, does not exceed about '003 of the breadth; while in uneasy ships it is sometimes as much as about '04 of the breadth.

II. *Dipping produced by Rolling* arises from the upright water-section and the several inclined water-sections not being at the same perpendicular distance from the ship's centre of gravity; so that the centre of gravity, and with it the whole ship bodily, is compelled to move vertically (generally to rise) when the ship heels over, as has been explained in Chapter III., Article 112. When such is the case, each *single roll* of the ship sets up a series of dipping oscillations, which go on until they are gradually stopped by the resistance of the water. Should the periodic time of dipping be either nearly or exactly half of the period of rolling, the extent of the alternate dipping and rising goes on increasing with each successive roll, until the tendency to augment it is balanced by the resistance of the water, and it then becomes permanent as long as the rolling continues; and such is the principal cause of "uneasy rolling."

The better to explain the difference between easy-rolling and uneasy-rolling vessels, and the nature of the motion of each of

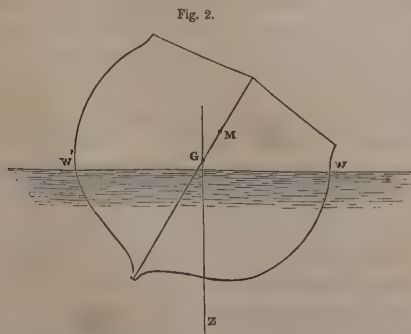


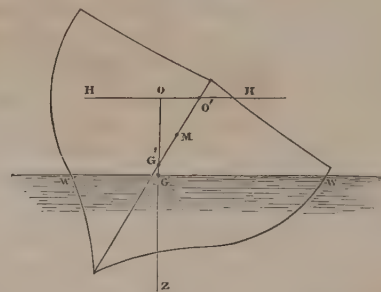
Fig. 2.

classes of figures, in order that their peculiarities may be the more distinct. In each of the figures, WW' represents the surface of the water, GZ, a vertical line through the centre of gravity of the ship, and M, her metacentre.

In the *Easy-Rolling Ship*, Fig. 2, the cross sections between wind and water are supposed to be of such figures that a longitudinal axis traversing the centre of gravity, G, is itself the *axis of level motion* (Art. 113). The load-water-sections, upright and inclined, are all at exactly the same perpendicular distance from the centre of gravity; and the centre of gravity remains quite immovable while the ship rolls.

In the *Uneasy-Rolling Ship*, Fig. 3, the cross sections between wind and water are supposed to flare out in such a manner, that the axis of level motion, from which the upright and inclined water-sections are equally distant, as explained in Art. 113, is at a considerable height above the centre of gravity. In the upright position of the ship, G represents the centre of gravity, and O the

Fig. 3.



axis of level motion; and HH is a horizontal plane traversing that axis. When the ship heels over into the inclined position shown in the figure, the axis of level motion shifts sideways from O to O', keeping at an uniform height above the surface of the water; and the centre of gravity rises vertically from G to G', the distance, $\overline{G'O'} = \overline{GO}$ remaining invariable. Such is the motion produced by a single roll in an uneasy ship.

But during a long series of rolls, the extent of the dipping oscillations, as already stated, gradually increases until it reaches a maximum, which is the greater, the more nearly the period of dipping approximates to *one half* of the period of rolling; and that maximum extent of vertical motion may greatly exceed the height, $\overline{GG'}$, shown in the figure. In such cases the axis, O, no longer moves to and fro horizontally, but acquires a vertical oscillation also; and the time of greatest elevation of the centre of gravity no longer coincides with the instant of greatest heel, but takes place nearly a quarter of a period later. The consequence is, that every particle in the ship, situated above or below her centre of gravity, describes a curve like a figure of eight, laid on its side (thus, ∞), or placed obliquely; and a mass situated above the centre of gravity, such as a heavy body on deck, moves in such a curve in the following manner:—During the middle half of each roll of the ship, the mass plunges obliquely downwards, in the direction towards which the ship is heeling, until she is approaching her greatest angle of heel, when it is suddenly tossed upwards, to begin another oblique downward plunge in the opposite direction. Some persons have tried to convey an idea of this sort of motion, by calling it "transverse pitching."

* The theory of this sort of motion is given in a paper by the Editor of this treatise, communicated to the Institution of Naval Architects in 1864.

them, the cross sections, Figs. 2 and 3, are given. They are extreme and somewhat exaggerated illustrations of those opposite

It may be observed, that in the easy-rolling ship, Fig. 2, the statical stability (Art. 116) is obtained wholly by *depressing the centre of buoyancy* in heeling; whereas in the uneasy-rolling ship, Fig. 3, it is obtained partly by *raising the centre of gravity*.

The methods of determining by measurement and calculation, whether a design for a ship fulfils the conditions of easy-rolling, have been fully explained and exemplified in Chap. III., Section IV., Articles 111, 112, 113, and 117.

III. *Dipping produced by Pitching* occurs when the centre of flotation and centre of buoyancy are not situated in the same vertical cross section, as explained in Chapter III., Article 120. When the periodic times of free dipping and pitching are not very different, the effect of this is very nearly the same as if the pitching oscillations, instead of being performed about a transverse axis through the centre of gravity, were performed about a transverse axis in the same vertical plane with the centre of flotation (F, Fig. 15 of Chap. III.), and at the same level with the centre of gravity; the square of the longitudinal radius of gyration being at the same time increased by a quantity equal to the square of the horizontal distance between the centre of gravity and centre of flotation (H F, in the same figure). This sort of motion constitutes *uneasy pitching*. Its cause and prevention have been sufficiently explained in the Article already referred to, and the rest of the 5th Section of Chap. III.

It is obvious that there is one very simple method of avoiding all the causes of uneasy motion described in this Article—viz., *to make the external figure of the ship, between wind and water, symmetrical about three planes at right angles to each other, traversing her centre of gravity, horizontally, longitudinally, and transversely*; for then the axis of level motion must necessarily traverse the centre of gravity; and the centres of immersion, emersion, and flotation, must all be in the same vertical cross section with the centre of buoyancy and centre of gravity. That method, however, is seldom followed in practice.

SECTION II.—OF THE MOTION OF WAVES.

134. *Wave-Motion in general described.* The transmission of a wave through a mass of fluid consists in the travelling onward of a certain shape of its surface and arrangement of its particles. The particles themselves do not travel permanently onward, but in rolling waves they oscillate or revolve about centres which are not far from their respective positions of repose; and in "waves of translation" they perform curved movements of limited extent.

The most obvious appearance presented to the eye by the waves of the sea is the onward travelling of a series of parallel ridges and furrows which are of equal or nearly equal size, and of a serpentine shape in outline. The tops of the ridges are called *crests*; the hollows between them, *troughs*; and when the trough of a wave is spoken of as having a definite level and position, the lowest part of the hollow is meant.

The *height* of a wave is measured from trough to crest. The greatest height which has yet been accurately measured in the open sea, is about 30 feet.

The level of still water is not exactly midway between the crests and the troughs; for the crests, being of a more peaked form than the troughs, rise somewhat more above that level than the troughs sink below it, as will afterwards be more particularly stated.

By the *length* of the waves of a series is meant the distance, measured in the direction of advance, from crest to crest, or from trough to trough. The greatest length which has yet been measured is about 500 feet.

It is sufficiently evident from the appearance of the surface of the water, that its particles, during wave-motion, move up and down through a vertical distance equal to the height of a wave. An inspection of the movements of floating bodies shows further, that the particles also move backward and forward through a horizontal distance, which is in some cases equal to, and in other cases greater than, the extent of their vertical motion; and that those movements are combined in such a manner, that each particle of water revolves in a vertical plane, in an orbit which in deep water is a circle, and in shallow water a flattened oval. The centre of the orbit is somewhat above the position of the particle in still water. A particle in the trough of a wave is moving backwards; on the front slope, upwards; on the crest, forwards; on the back slope, downwards.

The velocity of revolution and dimensions of the orbits of the successive layers of particles below the surface diminish in going downwards according to a law which will be afterwards stated.

If the whole mass of water, when still, be conceived to be divided into horizontal layers, and also into vertical columns, the serpentine rolling of each layer, when agitated by waves, may be compared to that of a sheet laid upon the ground and shaken up and down at one edge. The motion of the columns may be compared to the bending and swaying of the stalks in a wind-swept field of corn; with this addition, that each column of fluid becomes alternately taller and slenderer and shorter and thicker; being taller and slenderer while it is bending forwards, shorter and thicker while it is swaying back.

No wave can be continuously transmitted in which the height exceeds a certain proportion to the length. As that proportion is approached, the crests of the waves become gradually steeper and sharper, until at length they break up into foam and spray, or are made by the wind to curl over and fall forward.

Two or more series of waves may traverse the same mass of water in the same or different directions at the same time; the motion of each particle being the resultant of the several motions which it would have received from the several series of waves separately.

Waves undergo reflection at the face of a steep cliff or wall, and refraction (or change of direction) when passing from one depth of water to another. The figure of the free surface of a series of waves, as well as that of each *surface of equal pressure* by which the mass of agitated water may be conceived to be divided into layers, is governed by the general principle, that in a mass of fluid *each surface of equal pressure is perpendicular to the direction of the resultant pressure exerted by each particle that it traverses upon the neighbouring particles*. In a mass of fluid at rest, that direction is everywhere vertical, and the surfaces of equal pressure are all horizontal. In a mass of moving fluid, the resultant pressure exerted by a particle is compounded of the force of gravity, and of the re-action exerted by the particle in consequence of its motion (Article 76).

Although many questions as to the motion of waves remain to be cleared up by experiment and by mathematical investigation, the laws of the motions of the more frequent kinds of waves,

especially those of the ordinary ocean-swell, are known with great completeness and precision, theory having in every case been exactly confirmed by experiment. Those laws have an important bearing, not only on the oscillations of ships, but on their resistance and propulsion; and they therefore require to be stated somewhat fully, and illustrated by tables. For the details of mathematical and experimental investigations on this subject, reference may be made to Mr. Airy's article on *Tides and Waves* in the *Encyclopædia Metropolitana*; Mr. Scott Russell's researches in the Reports of the British Association for 1844; and to papers by Mr. Stokes in the Cambridge Transactions for 1842 and 1850; by Mr. Earnshaw in the Cambridge Transactions for 1845; by Mr. Froude in the Transactions of the Institution of Naval Architects for 1862; and by the Editor of this work in the Philosophical Transactions for the same year.

135. *Rolling Waves in Deep Water.* By "deep water," in discussing the motion of waves, is to be understood water so deep, that the nearness of the bottom has no sensible influence on the form of the particles' orbits, or on the length of the wave which travels with a given velocity. What degree of smallness

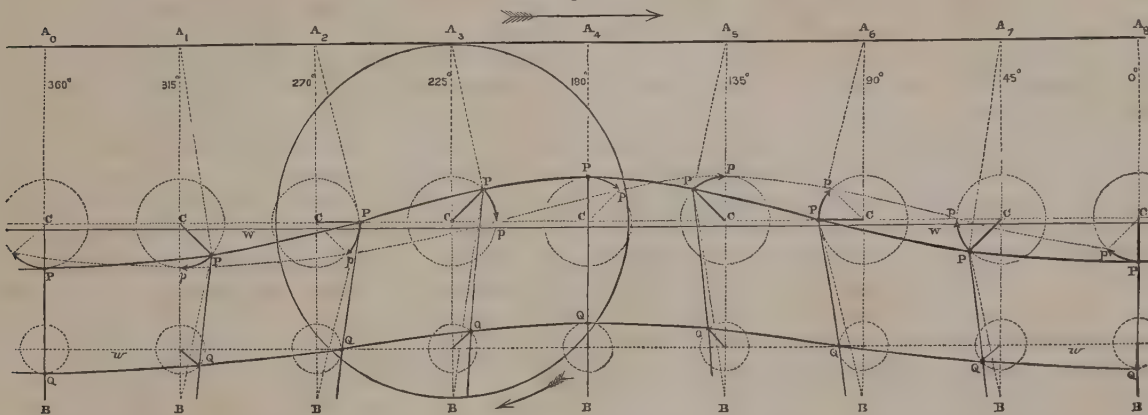
of such influence is to be regarded as insensible, depends on the purpose to which the results of the investigation are to be applied.

If, for example, we regard *one per cent.* as the smallest alteration in the length of a wave of which it is necessary to take account in connection with the motion of ships, it follows that all water of greater depth than that which produces that alteration is to be regarded as *deep water* for our present purpose; that is to say (as will appear in a later Article), all water whose depth is greater than five-twelfths of the length of a wave.

It is known by theory and by observation, that the particles of deep water affected by wave-motion revolve in vertical orbits which are exactly or almost exactly circular. Fig. 4 represents the mode of transmission of that motion, and contains one complete wave-length, extending from the bottom of one trough to the bottom of the next, a crest being in the middle.

The arrow at the top of the figure indicates the direction in which the waves are travelling. The horizontal line just below the arrow serves as a scale of distances: from A_0 to A_8 is a wave-length, which is divided into eight equal parts by the points $A_1, A_2, \&c.$ The height of that line above the centres

Fig. 4.



of the orbits of the surface particles depends upon principles which will be stated further on.

The upper plain wave-line marks the surface of a wave, at the instant when the crest is passing below the point A_4 , and the two troughs below the points A_0 and A_8 . The points marked $P, P, P, \&c.$, represent nine particles in that surface, which, when the water is still, are equidistant from each other, and situated in the level line $W W$, directly below the nine points $A_0, A_1, \&c.$ The orbits of those particles are represented by nine dotted circles, described about the nine centres marked C .

After the lapse of one-eighth of a wave-period, each particle has moved through one-eighth of a revolution, or 45° of its orbit, as shown by the short curved arrows. The points of those arrows represent the new positions of the particles; and the dotted wave-line traversing those points shows the new position of the wave, which has travelled forward through one-eighth of a wave-length; its crest being now below the point A_5 , the following trough below A_1 , and the preceding trough below a point one-eighth of a wave-length in advance of A_8 .

The lower plain wave-line represents the first position of a sub-surface of equal pressure, which, when the water is still, coincides with the level dotted line $w w$. The nine points marked

Q represent nine particles in that surface; their orbits are represented by small dotted circles; and their motions, although performed in smaller orbits, exactly correspond and keep time with those of the particles marked P , which are respectively above them.

The slightly curved lines marked $P Q$, represent *originally vertical* columns of particles, and show how those columns bend and sway with the wave-motion so as at any given particle to lean in the opposite direction to that in which the wave-surface slopes. The curved and distorted quadrilateral figures inclosed between them and the wave-lines, show the shapes successively assumed by a series of equal and *originally rectangular* blocks of water.

The shape of the wave-surfaces, and the velocity of transmission of the waves, depend on the following principles:—

From C , the centre of the orbit of any particle P , draw CA vertically upwards, and of the length given by the following proportion—

$$\begin{aligned} &\text{As the centrifugal force of } P \\ &: \text{is to gravity} \\ &: : \text{so is } CP \\ &: \text{to } \overline{CA}; \end{aligned}$$

that is to say (agreeably to Chapter II., Articles 81, 82), make \overline{CA} equal to the height of the revolving pendulum whose periodic time is the same with that of the wave-motion.

The pressure with which the particle P reacts on the neighbouring particles is the resultant of gravity, represented by AC, and centrifugal force, represented by CP; that pressure, therefore, is represented by \overline{AP} ; which line, therefore, is a normal to the wave-surface.

Hence it follows, that if a circular disc, of the radius \overline{CA} , with a tracing point, P, fixed in it, be rolled along the under side of the straight line A_0, A_0 , the trochoid traced by the point P will be the figure of the wave-surface; and the wave-length will be equal to the circumference of the rolling circle.

The period, therefore, of a wave of a given length in deep water, is the same with that of a revolving pendulum whose height is to the length of the wave as the radius of a circle to the circumference. The velocity of advance of the wave is found by dividing the length of the wave by its period.

The velocity of the particle P is to the velocity of advance of the wave, as the tracing-arm, \overline{CP} , is to the radius of the rolling circle, \overline{CA} .

For each sub-surface of equal pressure, such as that traversing the particles marked Q, the rolling circle is the same as for the upper surface; it is only the tracing arm that is smaller.

By reasoning for which reference must be made to the authorities cited in the last Article, it is proved, that the orbits of the particles diminish in geometrical progression in going downwards; and that the inclination of the originally vertical column at any particle is found by the following construction:—From the lower end, B, of the vertical diameter, BA, of the rolling circle, draw the straight line BP to the particle in question; that line will be a tangent to the curve, PQ, of the originally vertical column at P.

From this construction it appears, that in waves of moderate height compared with their length, the inclination of the originally vertical column of water at a given particle, is nearly, though not exactly, equal to the slope of the wave-surface.

136. Rules and Tables for Waves in Deep Water.—

I. Height of equivalent pendulum (or radius of rolling circle), in feet, = $\cdot 8154 \times$ square of period in seconds.

II. Length of wave in feet = $6\cdot 2832 \times$ equivalent pendulum
= $5\cdot 1233 \times$ square of period in seconds

$$= \frac{\text{square of velocity in feet per second}}{5\cdot 1233} = \text{velocity} \times \text{period.}$$

III. Velocity in feet per second = Period $\times 5\cdot 1233$

$$= \sqrt{\{32\cdot 2 \times \text{equivalent pendulum}\}}$$

$$= \sqrt{\{5\cdot 1233 \times \text{length of wave}\}}$$

$$= \text{Length} \div \text{Period.}$$

IV. Velocity in knots = Velocity in feet per second $\div 1\cdot 688$.

V. Velocity in statute miles an hour = Velocity in knots $\times 1\cdot 151$.

VI. Sine of steepest slope of surface = $\frac{\text{Height}}{2 \times \text{Equivalent Pendulum}}$

$$= \frac{3\cdot 1416 \times \text{Height}}{\text{Length of Wave}}$$

(The angle of slope in degrees is roughly equal to $\frac{180 \text{ height}}{\text{length}}$.)

VII. Velocity of particles of water at surface

$$= \text{Velocity of wave} \times \text{sine of steepest slope of surface.}$$

(The height from trough to crest is the diameter of the orbits of the particles of water.)

VIII. To find the ratio in which the orbits and velocities of the particles are diminished at a given depth below the surface:—Divide the given depth by the equivalent pendulum; the natural number answering to the quotient in a table of hyperbolic logarithms will be the reciprocal of the ratio required.

The following approximate rule is very nearly correct:—

The orbits and velocities of the particles of water are diminished by one-half, for each additional depth below the surface equal to one-ninth of a wave-length. For example—

Depths in fractions of a wave-length: $0 \frac{1}{9}, \frac{2}{9}, \frac{3}{9}, \frac{4}{9}$, &c.

Proportionate velocities and diameters: $1 \frac{1}{2}, \frac{1}{4}, \frac{1}{8}, \frac{1}{16}$, &c.

IX. To find how high the centre of the orbit of a given particle is above the level of that particle in still water:

Divide the square of the diameter of the orbit by eight times the equivalent pendulum of the waves; or—

Divide the square of the velocity of the particle, in feet per second, by $64\cdot 4$, for the height in feet.

X. To find the mechanical energy of a layer of water agitated by wave-motion: multiply the weight of the layer by twice the height at which the centres of the orbits of its particles stand above the positions of those particles when the water is still.

One half of this energy consists in motion, and the other half in elevation.

XI. To find the mechanical energy of a mass of water of a given horizontal area and unlimited depth, agitated by waves: multiply the area by one-sixteenth part of the square of the height of the waves, and by the heaviness of the fluid (64 lbs. per cubic foot, for sea-water).

XII. To find the mechanical energy of one wave-length of a layer of water of a given breadth and thickness: multiply together the breadth and thickness of the layer, the square of the diameter of the orbits of the particles in it, the heaviness of the fluid, and the constant, $1\cdot 5708 \left(\frac{\pi}{2}\right)$.

TABLE OF PERIODS AND LENGTHS OF WAVES IN DEEP WATER, ARRANGED ACCORDING TO THEIR VELOCITIES IN KNOTS.

Velocity. Knots per hour.	Velocity. Feet per second.	Velocity. Statute Miles per hour.	Period. Seconds.	Equivalent Pendulum. Feet.	Length. Feet.
1	1·688	1·15	0·33	0·09	0·56
2	3·376	2·30	0·66	0·36	2·25
3	5·064	3·45	0·98	0·80	5·06
4	6·752	4·60	1·31	1·43	9·00
5	8·44	5·75	1·64	2·24	14·05
6	10·13	6·91	1·97	3·22	20·2
7	11·82	8·06	2·30	4·38	27·5
8	13·50	9·21	2·63	5·72	36·0
9	15·19	10·36	2·96	7·24	45·5
10	16·88	11·51	3·29	8·94	56·2
11	18·57	12·66	3·62	10·8	68·0
12	20·26	13·81	3·95	12·9	80·9
13	21·94	14·96	4·27	15·1	95·0
14	23·63	16·11	4·60	17·5	110·1
15	25·32	17·26	4·93	20·1	126·4
16	27·01	18·42	5·26	22·9	143·8
17	28·70	19·57	5·59	25·8	162·3
18	30·38	20·72	5·92	29·0	182·0
19	32·07	21·87	6·25	32·3	202·8
20	33·76	23·02	6·58	35·8	224·7
21	35·45	24·17	6·91	39·4	247·8
22	37·14	25·32	7·24	43·3	272·0
23	38·82	26·47	7·57	47·3	297·3
24	40·51	27·62	7·90	51·5	323·6
25	42·20	28·77	8·23	55·9	351·2
26	43·89	29·93	8·56	60·4	379·8
27	45·58	31·08	8·89	65·2	409·6
28	47·26	32·23	9·21	70·1	440·5
29	48·95	33·38	9·54	75·2	472·5
30	50·64	34·53	9·87	80·5	505·7

137. *Rolling Waves in Shallow Water.*—In shallow water of uniform depth, the orbit of each particle becomes an oval of a height less than its length, and nearly, but not exactly, of an elliptical form. The oval orbits are smaller, and also more flattened, the nearer the particles are to the bottom: the particles in contact with the bottom oscillate back and forward in straight lines. The length and the velocity of advance of a wave of a given period are less in shallow than in deep water; and in order that a wave may advance with a given speed in shallow water, it must be of greater length and longer period than a wave which advances with the same speed in deep water.

When water is *very* shallow compared with the length of the wave, the velocity of advance becomes sensibly independent of the length and period of the wave, and is nearly equal to the velocity acquired by a heavy body in falling through one half of the depth from the surface to the bottom (Article 74): also, the extent of horizontal displacement is nearly the same from the surface to the bottom; and the extent of the vertical displacement of any particle is nearly proportional to its mean height above the bottom.

In cases in which the depth of the water is such that both it and the length of the wave have a material influence on the velocity of advance (that is to say, speaking roughly, when the depth of the water is between one thirty-sixth and five-twelfths of the length of a wave), that velocity follows a complex law which cannot be clearly expressed except by algebraical symbols. Rules for its application, however, will be added in words, and illustrated by a few examples arranged in a Table.

[In algebraical symbols,

Let L denote the length of a wave, in feet;

T , its periodic time;

V , its velocity of advance, in feet per second;

h , the depth from the centres of the orbits of the surface particles to the bottom;

g , gravity (32.2 feet per second);

e , the base of the Naperian logarithms;

a , the horizontal semiaxis, and b , the vertical semiaxis, of the orbit of a surface particle;

Then,

$$\frac{b}{a} = \frac{e^{\frac{2\pi h}{L}} - e^{-\frac{2\pi h}{L}}}{e^{\frac{2\pi h}{L}} + e^{-\frac{2\pi h}{L}}}; \dots \dots \dots (1.)$$

$$V = \sqrt{\frac{g L b}{2\pi a}}; \dots \dots \dots (2.)$$

$$T = \sqrt{\frac{2\pi L a}{g b}} = \frac{L}{V}; \dots \dots \dots (3.)$$

The following rules express those equations:—

I. To find the ratio of the height of a surface particle's orbit to the breadth of that orbit: divide the depth from the centre of that orbit to the bottom by the radius of a circle whose circumference is the length of a wave; find the number of which the quotient of that division is the hyperbolic logarithm, and also the reciprocal of that number; then—

As the sum of the number and its reciprocal

: is to their difference

: : so is the breadth of the orbit

: to its height.

II. To find the velocity of advance of a wave of a given length: compute its velocity as if for deep water, by Rule III.

of Article 136; and multiply that velocity by the square root of the ratio in which the height of a surface particle's orbit is less than the breadth.

III. To find the period of the same wave: divide the length by the velocity.

IV. The length of the wave which travels with a given speed in shallow water is greater than that of the wave which travels with the same speed in deep water, in the ratio of the breadth to the height of a surface particle's orbit.

V. The length and velocity of a wave of a given period are less in shallow than in deep water, in the proportion of the height to the breadth of a surface particle's orbit.

VI. To find the velocity of a wave in *very* shallow water, in feet per second: multiply the depth in feet from the centres of the surface particles' orbits to the bottom by 32.2, and extract the square root of the product.

VII. To find the length, in *very* shallow water, of a wave of a given period in seconds. Find the length corresponding to the given period in deep water, by Rule II. of Article 136; and take a mean proportional between that length and the circumference of a circle whose radius is the depth of the water.

(Rules VI. and VII. are applicable with some precision, where the depth of the water does not exceed about $\frac{1}{18}$ of the length of the actual wave; and in a rough way, when it does not exceed $\frac{1}{18}$.)

TABLE OF EXAMPLES.

Depth from centres of orbits of surface particles to the bottom, in fractions of a wave-length.	Ratios of Quantities for Shallow Water to the corresponding quantities for Deep Water.		
	Velocity for a given Length.	Length and Velocity for a given Period.	Length for a given Velocity.
$\frac{1}{36}$417	.174	5.76
$\frac{1}{30} = \frac{1}{18}$.579	.386	2.98
$\frac{1}{24} = \frac{1}{12}$.693	.481	2.08
$\frac{1}{20} = \frac{1}{10}$.776	.603	1.66
$\frac{1}{18}$838	.708	1.42
$\frac{1}{16} = \frac{1}{8}$.884	.781	1.28
$\frac{1}{14} = \frac{1}{7}$.940	.884	1.13
$\frac{1}{12} = \frac{1}{6}$.969	.939	1.06
$\frac{1}{10} = \frac{1}{5}$.985	.970	1.03
$\frac{1}{9} = \frac{1}{9}$.995	.989	1.01

138. *Surf-Waves and Breakers.* When waves travel from deep into shallow water, their periodic time continues unaltered; and therefore they become gradually shorter, according to the principles stated in Article 137. The energy of each wave is thus gradually communicated to a smaller and smaller mass of water, and the extent of motion of the particles consequently increases. Thus it is that waves are produced, on some rocky coasts, of 150 feet high. But to that increase a limit is put by the breaking of the wave-crests (already referred to in Art. 134), which begins to take place when the horizontal semiaxis of the orbits of the surface particles is approaching to a length equal to the radius of a circle whose circumference is a wave-length. Thus the waves travelling over long shallows are gradually broken down, and their energy expended in producing foam and eddies.

Another cause of the breaking of waves in travelling over long shallows is, that when the height of the waves bears any considerable proportion to the depth of the water, the crests travel faster than the troughs; so that the front of each wave becomes by degrees steeper than the back, and at length curls forward and falls over.

139. *Waves of Translation* are produced by the transmission of a motion of such a kind, that each particle, instead of returning periodically to the point from which it set out, is brought, at the

end of the wave-period, to a point in advance of or behind its original position as the case may be. The existence and phenomena of waves of this class were discovered by Mr. Scott Russell.* A wave of translation is said to be positive or negative, according as it shifts the particles of water forwards or backwards.

A positive wave of translation is *solitary*; that is, it travels alone, and is not necessarily preceded or followed by a series of other waves. Its surface presents a swell rising above the level of still water throughout its whole length; and each particle at which it arrives is lifted up from a state of rest, carried forward in an arched curve, and set down at rest in a position at a certain distance in advance of its original position, which distance is sensibly the same throughout the whole depth of the water. The length of a positive solitary wave of translation is shorter, by the distance through which a particle is transferred, than the circumference of a circle whose radius is the undisturbed depth of the water. The velocity with which the wave travels is sensibly equal to that acquired by a heavy body in falling through half the depth from the *crest* of the wave to the bottom of the water.

If a wave of translation is a *negative* wave, or wave of backward translation, it presents a hollow, sinking below the level of still water throughout its whole length; and each particle at which it arrives is depressed from a state of rest, and carried backward in an inverted arched curve. A negative solitary wave is longer, by the distance through which a particle is transferred, than the circumference of a circle whose radius is the undisturbed depth of the water; and it travels with a velocity equal to that acquired by a heavy body in falling through half the depth from the *trough* of the wave to the bottom of the water.

The height of the *equivalent pendulum* of a wave of translation, is equal to the depth from the crest or trough of the wave, as the case may be, to the bottom of the water.

In all Mr. Scott Russell's experiments on negative waves of translation, each such wave was followed by a series of rolling waves; but it is not inconceivable that a solitary negative wave of translation might be produced by some special mode of experimenting.

From what has been stated respecting the velocity and length of a wave of translation, it appears that such a wave, if positive, is shorter, and if negative, longer, by the extent of the translation, than the rolling wave which travels in *deep water* with the same speed; and when the height of the wave, and consequently the extent of translation, are small compared with its length, the relations between the velocity, length, and period of the wave of translation are approximately the same with those given for rolling waves in deep water, by Rules I., II., III., IV., and V., of Article 136, and by the Table at the end of that Article.

A *wave of translation in deep water* may be conceived to be formed *approximately* as follows:—With the revolving motion of the particles in a rolling wave, let there be combined, if the wave is positive, a forward current in each layer of water, and if the wave is negative, a backward current in each layer of water, having a velocity equal to the velocity of the revolving motion. The wave thus produced will have the same relations between its velocity, length, and period, with the wave of translation in shallow water already described, and the form of its surface will be nearly

the same; but the extent of translation will diminish, in going downwards, in geometrical progression.

The *bore* (a species of tidal wave in shallow water) may be regarded as consisting of the front half of a very steep solitary positive wave of translation, advancing into still water, and followed by a permanent elevation of the water, and by a current whose velocity is that of the particles of water at the top of their arched paths.

SECTION III.—OSCILLATIONS OF A SHIP AMONGST WAVES.

140. *General Explanations.*—The waves whose effects upon the oscillations of a ship it is most important to consider, are the rolling waves of deep water, described in Articles 135 and 136, and represented in Fig. 4 of this chapter.

If a ship were wholly without stability, her centre of gravity, centre of buoyancy, and metacentre coinciding in one point, the motion assumed by that point would be exactly that of the centre of gravity of the mass of water displaced by the ship; that is to say, it would revolve once in each wave-period in a vertical circle, of the same diameter with the orbits of the particles of water situated in the same layer. Such is the motion which the centre of buoyancy of a ship tends to perform amongst waves, and does actually perform in the absence of forces tending to modify that motion; and the same statement, subject to the same condition, applies to the centre of gravity, and to every other point in the ship.

To this motion of the ship it is proposed to give the name of *passive heaving*, that term being understood to comprehend the swaying from side to side, as well as the rising and sinking, of which the orbital motion is compounded. It will be seen in the sequel that the passive heaving motion is modified by the progression of the ship when under way, and that it gives rise to a swerving to and fro, or oscillation about a vertical axis, called *yawing*.

The weight of the ship being combined with the centrifugal force due to her heaving motion, gives a resultant reaction through her centre of gravity, inclined to the vertical in a direction which, for passive heaving, is perpendicular to the wave-surface traversing the ship's centre of buoyancy (a surface which may be called the *Effective Wave-surface*); and that direction is the *apparent* direction of gravity on board the ship, as indicated by plumb-lines, pendulums, suspended barometers and lamps, spirit-levels, and the positions assumed by persons walking or standing on deck. The equal and opposite resultant pressure of the water, acting through the centre of buoyancy, is in like manner compounded of actions due to weight and centrifugal force; and it acts in a line normal to the effective wave-surface; that is, parallel to the resultant reaction of the ship. Those two forces balance each other, not when the ship's upright axis is vertical, but when it is normal to the effective wave-surface; and when she deviates from that position, they form a righting couple tending to restore her to it. Thus the stability of a ship amongst waves, instead of tending to keep her steady, as in smooth water, tends to keep her *upright to the effective wave-surface*; and such is the motion of any vessel or other floating body having great stability and small inertia, such as a light raft. This may be called *passive rolling*, or *rolling with the waves*.

Passive rolling is modified by the inertia of the ship, which makes her tend to perform oscillations in the same periodic time as in still water, by the impulse and resistance of the particles of

* Reports of the British Association, 1844. An investigation of the theory of those waves by Mr. Earnshaw, appeared in the Cambridge Transactions for the same year.

water against her keel and the sharp parts of her hull—which tend, under certain circumstances, to make her roll *against* the waves, that is, inclining towards the nearest wave-crest—and by other circumstances.

The tendency to keep upright to the effective wave-surface may be distinguished from the tendency to keep truly upright, by calling the former *stiffness* and the latter *steadiness*. In smooth water, stiffness and steadiness are the same thing; amongst waves, they are different, and sometimes to a certain extent opposed: that is to say, the means used for obtaining one of those qualities are sometimes prejudicial to the other. Stiffness is favourable to the dryness of the ship, and to the power of carrying sail; steadiness is favourable to her strength and durability, and the safety of her lading; and in ships of war, to the power of working guns in rough weather.

A ship whose course is either oblique or transverse to the wave-crests is made by the waves to perform a series of longitudinal oscillations, which may be called *passive pitching and scending*.

In all the oscillatory movements which a ship performs amongst waves, two series of oscillations are combined—those in which the ship keeps time with the waves, being her passive or forced oscillations, and those which she performs in periods depending on her own mass and figure, as in smooth water, being what may be called her free oscillations. The tendency and ultimate effect of the resistance of the water is to destroy the free oscillations after a certain time; so that the forced oscillations alone are permanent.

The true principles of the rolling of a ship amongst waves were first set forth in a series of papers by Mr. Froude, in the Transactions of the Institution of Naval Architects for 1861, 1862, and 1863. The same subject is considered in a paper by Mr. Scott Russell in the volume of that publication for 1863; and certain special matters connected with it are investigated by the editor of this treatise in the volumes for 1862 and 1864.

141. *Passive Heaving*, or the motion of a ship when each of her particles performs an orbital motion similar and equal to that of a certain particle of the water in which she floats, takes place when the ship floats amongst waves without having progressive motion.

CASE I.—To find the diameter of the orbit in which each particle of the ship moves, when the extent of wave-surface covered by her is small compared with the length of a wave, it is sufficient to find the depth of her centre of buoyancy below the surface of the water, and diminish the diameter of the orbit of a surface particle in the proportion corresponding to that depth, according to Rule VIII. of Article 136.

For example, suppose a ship whose centre of buoyancy is 8 feet below her plane of flotation, to be floating passively amongst waves 144 feet long, and 10 feet high from trough to crest, with her broadside to the waves, so that the extent of wave-surface which she covers is small compared with the length of a wave. The depth, 8 feet, from the surface of the water to her centre of buoyancy, is $\frac{1}{18}$ th of the length of a wave; and at that depth, according to Rule VIII. of Article 136, the diameter of the orbit of a particle of water is $\sqrt{\frac{1}{18}}$ or $\cdot 7$, nearly, of the height of the wave; therefore, the extent of the passive heaving of the ship, or diameter of the orbit in which her centre of gravity revolves, is

$$10 \times \cdot 7 = 7 \text{ feet,}$$

being the height of the effective wave-surface.

CASE II.—But if the ship covers an extent of wave-surface which is considerable compared with the length of a wave

(as when, for instance, she rides with her head to the sea) her centre of gravity takes a motion at each instant which is the mean of the motions of the several parts of the effective wave-surface which she covers; and that mean motion is always less extensive than the motion found in Case I., in a proportion which may be found approximately as follows:—Find the length of wave-surface covered by the ship, in a direction parallel to that in which the waves travel, and multiply that length by her “coefficient of fineness” (Art. 97), for a *reduced length*. About a centre, C (Fig. 5), describe a circle, whose circumference represents one wave-length; and in that circumference take an arc, ADB, to represent the reduced length before mentioned. Measure the chord A B. Then the extent of the heaving motion of the ship’s centre of gravity will be less than the extent calculated as in Case I., in the following ratio:—

$$\frac{\text{Chord A B}}{\text{Arc A D B}}, \text{ nearly.}$$

For example, suppose the ship already mentioned to ride with her head to the sea; that her length is 180 feet, and her coefficient of fineness $\cdot 4$. Then the reduced length is 72 feet; and if the circumference of the circle in Fig. 5 corresponds to 144 feet, 72 feet will correspond to a semicircle; for which the ratio $\frac{\text{chord}}{\text{arc}}$ is $\cdot 637$. The extent of heaving in Case I. was 7 feet;

therefore in the present case it will be

$$7 \times \cdot 637 = 4\frac{1}{2} \text{ feet, nearly.}$$

This, it may be observed, is but a rough approximation; but it is near enough for the present purpose.

Amongst short waves, the arc, ADB, representing the ship’s length, may amount to more than one circumference. The rule, however, is still applicable.

Half the difference between the extent of heaving of the ship and the height of the waves is the extent to which, during the passage of the waves, her depth of immersion amidships is liable to be alternately increased above and diminished below her depth of immersion in smooth water. In the example of Case I., that half-difference is—

$$\frac{10 - 7}{2} = 1\frac{1}{2} \text{ foot;}$$

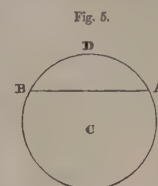
and in the example of Case II., it is—

$$\frac{10 - 4\frac{1}{2}}{2} = 2\frac{3}{4} \text{ feet.}$$

It appears, then, that deep immersion, and large horizontal dimensions, but especially deep immersion, tend to diminish the extent of the heaving motion of the ship as compared with that of the waves; and that the effect of those causes in producing such diminution is greatest amongst comparatively short waves.

142. *Relative Progression of Ships and Waves*.—The progression of the ship, when under way, alters the action of the waves upon her in various ways which will be afterwards explained, and which depend mainly upon the *apparent period* of the waves relatively to the ship—that is, the interval of time between the arrival of two successive crests, or two successive troughs, at the ship—and upon the *apparent slope* of the effective wave-surface in a direction athwart the ship (the latter circumstance being connected mainly with forced rolling oscillations).

The following is the geometrical construction for finding those two quantities:—



In Fig. 6, draw a straight line \overline{AU} to represent the direction and velocity of transmission of the waves; and in the same line, take \overline{AR} to represent their steepest actual slope. On AR as a diameter describe a circle.

From A draw a straight line to represent the direction and speed of the vessel's motion (such as AV , or AV' , according as that direction makes an acute or an obtuse angle with the direction of transmission of the waves). From the end of that line (V or V') let fall a perpendicular (VC or $V'C'$) on AU (produced if necessary). Then:—

I. The apparent velocity of the waves relatively to the ship is represented by CU (or $C'U$, as the case may be); and their *apparent period* is found by multiplying the true period by the inverse ratio of their apparent and real velocities, viz:—

$$\frac{\overline{AU}}{\overline{CU}} \text{ or } \frac{\overline{AU}}{\overline{C'U}}.$$

The *effective height of the equivalent pendulum of the waves*, is of course found by multiplying the actual height by the square of the same ratio. Hence it appears, that the effective period of the waves is greater or less than the actual period, according as the vessel's course lies away from or near the wind.

II. The thwart-ship slope of the waves is represented by the chord \overline{AS} (or $\overline{AS'}$, as the case may be) cut off by the circle AR from a line at right angles to that representing the course of the ship.

[In algebraical symbols

let u represent the velocity with which the waves travel;

v , the velocity of the ship;

ϕ , the angle which her course makes with the direction in which the waves are advancing; then

$$\text{apparent period of the waves} = \text{real period} \times \frac{u}{u \pm v \cos. \phi} \quad (1)$$

the sign $\left\{ \begin{matrix} - \\ + \end{matrix} \right\}$ being used according as ϕ is $\left\{ \begin{matrix} \text{acute} \\ \text{obtuse} \end{matrix} \right\}$;

$$\text{slope athwart-ship} = \text{actual slope} \times \sin. \phi, \text{ nearly.} \quad (2).$$

143. *Heaving modified by Progression.*—When the apparent periodic time of the waves is modified by the progressive motion of the ship, the time during which the forces act which produce the heaving motion of the ship is altered in the ratio of the apparent period to the true period; and the extent of the heaving motion is also altered in a proportion, which, for moderate deviations of the apparent from the true period, varies nearly as the square of that ratio. This law, however, does not continue to hold for a very great increase of the apparent period; the extent of heaving being less than the ratio just mentioned.

Hence the heaving motion of a ship is more extensive than that of the effective wave-surface, when the angle made by her course with the direction of advance of the waves is acute; and less extensive, when that angle is obtuse.

144. *Yawing*, or swerving of the vessel from side to side, by oscillation about her upright axis, is, when produced by the waves, the effect of the lateral swaying, which forms the horizontal component of the heaving motion, taking place with different velocities, or in opposite directions, at the bow and stern of the

vessel. The forces producing it are greatest when her course lies diagonally with respect to the direction of advance of the waves. It makes her fall off, or swerve from the wind, in mounting the crest of a wave, and come up, or swerve towards the wind again, in descending into the trough.

The extent of yawing oscillations is modified by the progressive motion of the ship, in the same manner with that of the heaving oscillations of her centre of gravity; and hence it may sometimes become very great, causing the ship to "steer wildly," through the lengthening of the apparent period of the waves when she is running nearly before them.

145. *Passive Pitching and Scending* are the longitudinal oscillations which a ship performs in descending and ascending the slopes of the waves. It is important to the safety of the ship that those movements should be performed in a lively manner, so that she may not ship seas over her bow or stern: in short, her longitudinal oscillations should accompany those of the surface of the effective wave, or nearly so.

The conditions upon which liveliness in pitching and scending depends have been explained in Article 130.

146. *Passive Rolling* is that kind of rolling oscillation in which the ship is forced to keep time with the waves.

It has already been explained in Article 140, that the weight of the ship being compounded with the centrifugal force due to her heaving revolution, gives a resultant reaction in a direction normal to the effective wave-surface, and equal and opposite to the resultant pressure of the water; and those two forces balance each other only when the ship is *upright to the effective wave-surface*, and form, in every other position of the ship, a sort of righting couple, tending to bring her to that position, exactly as if the force of gravity were represented by AP in Fig. 4 (Art. 135) instead of AC . Hence a very light and very stiff ship tends to float like a raft, rolling *with the waves*, and assuming at every instant the same slope with the effective wave-surface, as shown in Fig. 7 at A_0, A_1, A_2, A_3, A_4 .

In that figure, the arrow at the top shows the direction in which the waves travel; and the positions successively assumed by a ship are numbered in their order.

Let a board, having very little inertia and no stability, be placed so as to float upright in smooth water; then when the water is agitated by waves, that board will accompany the motions of the *originally upright columns of water*, (already described in Arts. 134, 135); that is to say, it will roll *against* the waves, inclining at every instant in a direction contrary to the effective wave-surface; as shown in Fig. 7 at B_0, B_1, B_2, B_3, B_4 .

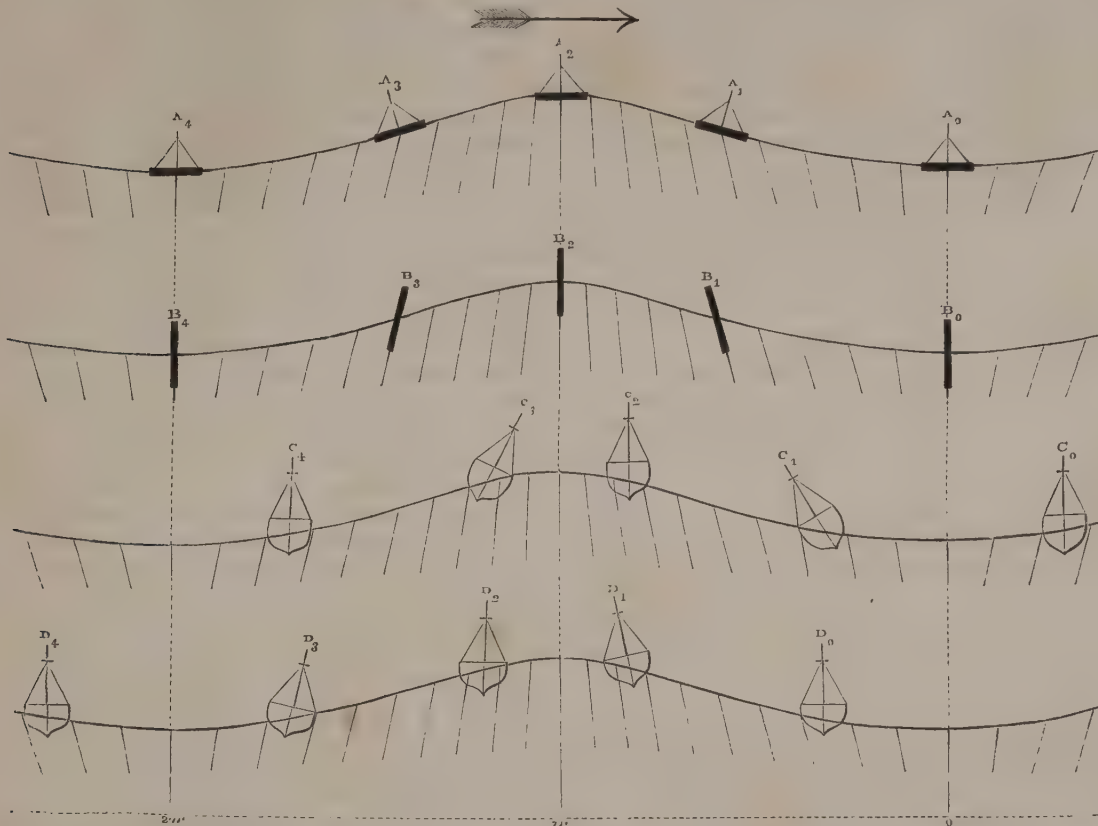
It has been shown by Mr. Scott Russell,* that the condition of the broad and rounded parts of a ship, and of her hull between wind and water, is analogous to that of the raft A ; while the condition of the keel, the sharp part of the floor, and the gripe and deadwood (or fine parts of the ends) is analogous to that of the board B ; so that the ship is under the action of two conflicting sets of forces: gravity, centrifugal force, and pressure, (constituting what may be called stiffness), tending to make her roll with the waves, like the raft A ; and the action of the water on the keel and the sharp parts of the hull (which may be called *keel-resistance*, as in Art. 128) tending to make her roll against the waves, like the board B ; and hence she will take some kind of intermediate motion.

* Transactions of the Institution of Naval Architects, 1863.

It has since been pointed out, however, by Mr. Froude and the editor of this treatise^{*} that there is an essential distinction between the two sets of forces before mentioned, in consequence of which, though conflicting, they are not directly opposed; viz.:—That the stiffness is an active force, which tends not only to prevent the ship from deviating from a position upright to the effective wave-

surface, but to restore her to that position after she has left it, with a force proportional to her deviation; while the keel-resistance is merely a passive force, opposing the deviation of the ship from the position of the originally vertical columns of water with a force depending, not on that deviation, but on the velocity of the relative motion of the ship and the particles of water, and not tending to

Fig. 7.



restore the ship to any definite position. Hence those two kinds of forces cannot directly counteract, but only modify, each other's effects.

For the mathematical investigation of the action of those forces, reference must be made to the already cited paper by the editor of this work, communicated to the Institution of Naval Architects in 1864. The following are the general conclusions arrived at:—

I. The permanent rolling of a ship of very great stability and without any sensible keel-resistance is governed by the motion of the effective wave-surface; so that she rolls *with the waves*, or like a raft, as shown in Fig. 7, A_0, A_1, A_2, A_3, A_4 .

II. When the period of unresisted rolling of the vessel is to the period of the waves as $\sqrt{2} : 1$, her permanent rolling is wholly governed by the motion of the originally vertical columns of water; so that she rolls *against the waves*, like a board of no stability floating edgewise, as shown in Fig. 1, B_0, B_1, B_2, B_3, B_4 .

In both of the preceding cases, the vessel is upright when the trough or crest of a wave passes her, as in the positions marked

0, 2, 4; and her angle of heel is equal to the steepest slope of the effective wave-surface.

III. When the period of unresisted rolling of the vessel is less than the above value, her upright positions occur *before* the arrival of the troughs and crests of the waves, as shown at C_0, C_2, C_4 ; and her angle of heel is *greater* than the steepest slope of the effective wave-surface, as shown at C_1, C_3 .

IV. The greatest angle of heel in permanent rolling occurs when the period of unresisted rolling of the ship is equal to that of the waves; and it exceeds the slope of the waves in a proportion which is the greater, the less the keel-resistance; and which becomes infinite when the keel-resistance vanishes.

V. When the period of unresisted rolling of the vessel is greater than $\sqrt{2} \times$ the period of the waves, her upright positions occur *after* the arrival of the troughs and crests of the waves, as shown at D_0, D_2, D_4 ; and her angle of heel is *less* than the steepest slope of the waves, as shown at D_1, D_3 .

147. *Passive Rolling—Geometrical Construction.*—The quantitative results of the investigation referred to in the preceding article can be represented by a geometrical construction, which will now be described.

^{*} Transactions of the Institution of Naval Architects, 1863-1864.

The data required are the following:—

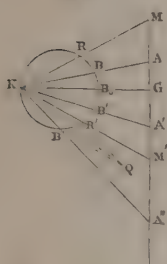
As to the Ship—

1. The height of her metacentre above her centre of gravity, which may be found by well-known methods;
2. Her radius of gyration, found as already described in Articles 126, 128;
3. Her leverage of keel-resistance, found as described in Article 128.

As to the Waves:—

4. The height of their equivalent pendulum, as to which see Art. 136, Rule I.; or, if the ship is under way, the height of the pendulum corresponding to the *apparent* period of the waves, as to which see Article 142.
5. The steepest slope of the effective wave-surface, as to which see Art. 136, Rule VI., and Art. 141; or when the ship lies obliquely to the waves, their steepest effective thwart-ship slope, as to which see Article 142.

Fig. 8.



Those five data are to be used in the following manner:—

In Fig. 8, draw a straight line MM' , in which lay off equal distances GM , GM' , both ways from G , to represent the height of the ship's metacentre above her centre of gravity.

Divide the square of the ship's radius of gyration, by the "equivalent pendulum" of the waves, and lay off MA to represent the quotient. MA may also be found by the following proportion:

As the square of the period of the waves,
: is to the square of the ship's period of free unresisted rolling,
: : so is the metacentric height MG ,
: to MA .

(MA in the figure corresponds to a vessel whose period is less than that of the waves; MA' , to a vessel whose period is greater than the period of the waves, and less than $\sqrt{2}$ times that period; MA'' , to a vessel whose period is greater than $\sqrt{2}$ times that of the waves. When the period of unresisted rolling is equal to that of the waves, A coincides with G ; when it is $\sqrt{2}$ times that of the waves, A coincides with M' .)

From G draw GK perpendicular to MM' , and of a length representing the "leverage of keel-resistance," and join KM , KG , KM' , KA .

On KM and KM' take equal distances, KR and KR' , to represent the steepest slope of the effective wave-surface; and through the three points, K , R , R' , describe a circle.

Then—

I. The angle, $M'KA$, is the *difference of phase* of the wave-motion and of the vessel's rolling; that is to say,

As a complete revolution,
: is to the angle $M'KA$,
: : so is the period of the waves
: to the time which elapses between the instant of the vessel's being upright, and the arrival of the crest or trough of a wave.

When A lies between M and M' , as at A or A' , the vessel becomes upright *before* the arrival of a trough or crest, as at C_0 , C_2 , and C_4 , in Fig. 7, referred to in Article 145, Case III. When A lies beyond M' , as at A'' , the vessel becomes upright *after* the

arrival of a trough or crest, as at D_0 , D_2 , and D_4 , in Fig. 7, referred to in Art. 145, Case V. When A coincides with M' , the vessel becomes upright at the instant of the arrival of each trough and crest, as at B_0 , B_2 , B_4 , in Fig. 7, referred to in Art. 145, Case II.

II. The chord, KB , cut off by the circle KRR' from KA , represents the angle of heel of the vessel. KB and KB' , each greater than the slope of the wave, KR , correspond to such cases as those mentioned in Art. 145, Case III., and represented by the ship C in Fig. 7; KB'' , less than KR , corresponds to Case V. of Art. 145, represented by the ship D in Fig. 7. KB_0 represents the maximum angle of heel, when the period of the ship is identical with that of the waves, as in Art. 15, Case IV.

III. To find the angle of heel of the ship *relatively to the effective wave-surface*, construct a parallelogram on KR' and KB ; its diagonal will represent the required angle. For example, in the case in which KB represents the absolute angle of heel of the ship, complete the parallelogram $KR'QB''$, and its diagonal KQ will represent the angle of heel of the vessel relatively to the surface of the waves. Upon that angle depends the straining action which the rolling produces on the frames at the bilges, and on the knees of the deck-beams.

148. *General Remarks on the Oscillations of Ships.*—The forced or passive oscillations of ships, described in the present Section, are those which produce the most severe strains, because of their continual recurrence; the free oscillations being gradually extinguished by the resistance of the water. It appears, however, that the periodic time of the free oscillations has an important influence on the extent of the forced oscillations, especially in rolling; the most unfavourable proportions of the periodic time of free rolling to that of passive rolling being those which lie near equality, and between equality and $\sqrt{2} : 1$; for the equality of these periods tends to produce excessively extensive rolling; and the ratio $\sqrt{2}$ to 1, and those near it, make the ship roll against the waves, thus throwing her into positions in which there is a risk of the wave-crests breaking into her.

A period of free rolling much less than that of passive rolling, gives great stiffness, and makes the ship accompany the motions of the effective wave-surface; a period of free rolling exceeding $\sqrt{2}$ times that of passive rolling is favourable to steadiness, provided (as explained in Art. 131) its length is produced by the inertia of the ship, and not by insufficient statical stability.

The action of the water on a deep keel, a sharp floor, or fine ends below water, tends to moderate the extent of rolling produced by coincidence, whether exact or approximate, of the periods of free and passive rolling; but at the same time it lessens the effect of a long period of free rolling in producing the same result.

A deep draught of water is favourable, on the whole, to steadiness, but not to stiffness.

Rolling the centre of gravity rise and fall relatively to the water in rolling, and the periodic time of the dipping motion so generated happen to be either exactly or nearly one half of that of the passive rolling motion, the result will be a very uneasy motion, like that described in Article 133.

The progressive motion of the vessel through the water promotes steadiness or unsteadiness, according as her course lies near or from the wind.

The steady pressure of the wind on the sails promotes steadiness, at a certain angle of heel depending on the moment of that

pressure; the sudden gusts of the wind produce lurching, according to the principles stated in Art. 129. Easy motion at the different angles of heel produced by the wind, is promoted by isochronism in free rolling (as to which see Art. 126A).

As to pitching, scending, and yawing, it is chiefly important that those oscillations should be performed in a lively manner amongst waves; and that object is best promoted by having the longitudinal radius of gyration short compared with the length of the ship.

Experiments on the oscillations of ships, in rough and smooth water, and especially in rough water, are much wanted. When made in rough water, it is essential that the extent of rolling and pitching should be found by observations of fixed objects, or of an instrument of the gyroscope class, like that invented by Professor Piazzi Smyth; and not by means of a pendulum, plummet, or spirit level; because, as explained in Article 140, these instruments show the direction, not of gravity, but of the resultant of gravity and centrifugal force, which is normal to the effective wave-surface, instead of being truly vertical. One of the results of this fact is,

that to a person who confines his attention to the positions of suspended objects, such as the barometer, and to his own sensations, the ship often seems to roll considerably, when a glance at the horizon or at the stars would show him that she is really steady.

It would appear that a very close approximation to the form and proportions which are most favourable to steadiness has in some cases been realized by practical trials alone; and that, independently of the steadying action of sails; for there are vessels which, when under steam alone, in any moderate swell, keep their decks very nearly parallel to the horizon. It is of great importance that the lines and dimensions, and distribution of the weights, of ships which have been found by experience to possess this excellent quality, should be carefully recorded for the information of naval architects.

On the other hand there are vessels (especially screw steamers) whose ordinary extent of rolling each way is from three to four times the slope of the waves.

CHAPTER V.

RESISTANCE, PROPULSION, AND MANŒUVRING OF SHIPS.

149. *The Subjects of this Chapter* are those already referred to in Chapter I., Articles 7 to 11: that is to say, the resistance opposed by the water to the motion of ships through it; the propulsion of vessels, whether by mechanism or by sails, so far as it is connected with the general design of the ship, (the details being reserved for two of the later Divisions of the treatise); and the means provided for manœuvring the ship, so far as they too are matters of general design. The Chapter is therefore divided into six Sections, as follows:—

Section I. explains the general laws of the resistance opposed by water to bodies moving through it, especially in so far as those laws are applicable to the resistance of ships.

Section II. relates to the adaptation of the forms and dimensions of ships to their intended speed, in order that no more resistance may have to be overcome than that which is unavoidable.

Section III. treats of the computation of the resistance of a vessel, and the power required to drive her, at a given proposed speed, and of her probable speed with a given power; and also of some questions regarding experimental trials of speed.

Section IV. explains certain general principles of the action of propelling apparatus, which require to be attended to in designing a vessel that is to be driven by the re-action of the water; a subject whose details belong to the sixth Division of the treatise.

Section V. explains certain general principles as to propulsion by sails; the details of which subject belong to the fifth Division of the treatise.

Section VI. relates to points in the general design of a ship which affect her "handiness," or manœuvring qualities.

150. *Temporary and Permanent Resistance.*—The resistance opposed by a fluid to the motion of a solid body through it may be distinguished into two parts, *temporary* and *permanent*.

The temporary resistance is that which takes place during acceleration only, and arises from its being necessary to set a certain mass of fluid in motion along with the solid: the permanent resistance is that which acts during the uniform motion of the solid. It is to permanent and not temporary resistance that the ensuing sections I. II. and III. specially relate.

SECTION I.—PERMANENT RESISTANCE OF WATER.

151. *Direct Actions of the Particles of Water—Pressure, Tenacity, Stiffness.*—The *direct* forces which the particles of water or of any other fluid exert, either in resisting each other's motion, or the motion of a solid body, such as a ship, are of three kinds: *Pressure, Tenacity, and Stiffness.*

I. *Pressure* is exerted by the particles of a fluid in a direction at right angles to the surface pressed upon. The *intensity* is expressed (as explained in Chapter II. Article 51) in units of force on the unit of area (as pounds on the square foot), or in units of height of an equivalent column of the fluid (such as feet of water). In a fluid in repose, the resultant of the pressures exerted by a particle in all directions on the neighbouring particles, is simply equal to the weight of the particle; and the intensity of the pressure at any point is equivalent simply to the weight of the column of fluid which is supported above that point. In a fluid in motion (if the effects of *tenacity* and *stiffness* are insensible), the resultant of the pressures exerted by a particle in all directions on the neighbouring particles is equal to the weight of the particle combined with the reaction due to its motion.

II. The *Tenacity* of a fluid is a force resisting the separation of its particles. Its chief visible effects consist in a toughness or tensile action of the external skin of a liquid mass, as mani-

fested in the phenomena of capillarity, such as the formation of drops, and the adhesion of the liquid to solid bodies.

III. The *Stiffness* of a fluid differs from that of a solid body in this—that the stiffness of a solid body is a force resisting the disfigurement and relative displacement of its particles as much when they are in repose as when they are in motion, whereas the stiffness of a fluid in repose is insensible; and that of a fluid in motion consists in a resistance to the relative motion of any two particles which are moving in the same, or parallel directions, with different speeds, and is proportional *directly to the difference of their velocities, and inversely to the distance between them.*^o

When the stiffness of a fluid acts directly in resisting the motion of a solid body through it, the resistance increases proportionally to the velocity of the solid body simply. When the pressure and adhesion so act, the resistance increases for some modes of action proportionally to the square of the velocity, and for other modes of action proportionally to higher powers of the velocity. Hence, although the direct action of stiffness is a comparatively large and important cause of resistance at low velocities, such as those of pendulums, of sluggish streams, and of small models of ships, it becomes, relatively to the direct action of pressure, a comparatively small and unimportant cause of resistance at high velocities, such as those of real ships. There is an *indirect* action of stiffness through the medium of pressure and adhesion, which continues to be an important cause of resistance at the highest velocities: it will be explained in a later Article.

152. *Use of a supposed Current.*—In reasonings and calculations respecting the resistance of ships, it is often convenient to conceive the ship as fixed, and the water as forming a current which flows past it with an uniform speed equal and opposite to the actual speed of the ship through the water. That supposition makes no alteration whatsoever in the relative conditions of the ship and the particles of water.

The impulse of actual currents upon fixed solid bodies has been found to be different from the resistance of still water to solid bodies moved through it. This is accounted for by the fact, that no actual current is in the condition of the supposed current just mentioned. The supposed current has all its particles moving steadily in parallel lines with equal velocities, except in so far as they are disturbed by the solid body; whereas in every actual current, the particles move with greater or less velocities according as they are more or less distant from the bed of the channel, and perform whirling movements in combination with their onward flow.

153. *Indirect Actions of Water on a Ship—Application of the Laws of Momentum and of Energy.*—According to what has been stated in Article 151, a great part of the resistance of the water to the motion of a ship through it is exerted by means of an indirect process; the immediate mode of action being by an increase of pressure against the bow as compared with the stern, but the original cause of that increase being stiffness or tenacity, or some change in the motions of the particles of water.

Under such circumstances, to compute the resistance to the motion of a ship through a determination of the pressures exerted directly upon the several parts of her immersed surface, would

involve processes of impracticable complexity; and the only means of solving the problem even approximately are indirect methods, founded upon one or other of two principles;—the *equality of impulse and momentum* (Chapter II. Article 71); and the *equality of energy and work* (Chapter II. Articles 64, 75).

The first of those principles is to be used when some definite known amount of *permanent* motion in the direction of the advance of the ship is communicated to a continually increasing quantity of water; for the *new momentum impressed on the water in that direction in each second* is equal to the backward reaction of the water, which acts as a resistance on the ship.

But in most cases the permanent motion impressed on the particles of water cannot be known until after the resistance has been determined, and then it is necessary to ascertain, by some means, the *work performed amongst the particles of water whilst the vessel advances through a lineal unit* (such as a foot); which quantity of work will be equal to the resistance.

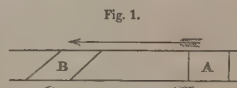
The processes amongst the particles of water, through which resistance to the ship's motion may be caused indirectly, may be thus enumerated:—

- The Distortion of the particles of Water;
- The production of Currents;
- The production of Waves;
- The production of Frictional Eddies.

Those causes of resistance will now be considered in succession.

154. *Resistance due to Distortion of the Particles of Water.*

Let Fig. 1 represent a mass of water whose particles are moving in parallel directions with different velocities, as indicated by the arrows.



Then any portion of the water which is rectangular at a given instant, as shown at A, undergoes distortion as it advances, as shown at B. To this kind of motion the particles oppose a passive resistance due to the *stiffness* already mentioned in Article 151; and that resistance is simply proportional to the speed with which the distortion goes on.

The particles of waves undergo alternate distortions of this kind in opposite directions; and so also do the particles of water in the neighbourhood of a moving ship.

The laws of this kind of resistance were investigated experimentally and theoretically by Mr. Stokes (Cambridge Transactions, 1850); and he showed, that although it forms the principal part of the resistance to small oscillations of pendulums, and an important part of the resistance which causes the gradual extinction of small waves (like those of the ripple raised by a light breeze), it becomes insensible in large waves, like the swell of the ocean, owing to the comparative slowness with which the molecular distortion goes on; and that conclusion is borne out by the fact, that the long waves of the Atlantic are known to travel 2000 miles, and upwards, with little diminution of size. Hence it is to be inferred that the resistance to distortion of the particles of water, although it may have a sensible effect on small models of vessels, is practically inappreciable in its action on actual ships; and this is a reason for great caution in applying to ships conclusions deduced from experiments on the resistance of models.

155. *Resistance due to the production of Currents.* When a stream of water has its motion modified in passing a solid body, and returns exactly to its original velocity and direction of motion

^o This law of the stiffness of fluids is conclusively proved by the researches of Mr. Stokes on the motion of pendulums in a resisting medium (Cambr. Trans. 1850), and by the experiments of Messrs. Humphreys and Abbot on the flow of the Mississippi (Report published by the U. S. Government).

before ceasing to act on the solid body, it exerts, on the whole, no resultant force on the solid body, because there is no permanent change of its momentum. But if, after the action between the stream and the solid body has ceased, the particles of the stream continue to follow the motion of the solid body, that body is resisted with a force equal to the forward momentum impressed on the fluid in each second. The following current thus produced is called the body's *wake*.

Every kind of resistance results in the production of more or less of a wake; but the special sort of wake considered in the present article is that which arises from the particles of water being *abruptly thrown off* from the solid body, and so made to cease their action upon it, at a time when they have a forward velocity bearing a definite relation to that of the solid.

For example, in Fig. 2, A represents a flat plate, propelled in a direction normal to itself through the water; and B represents a

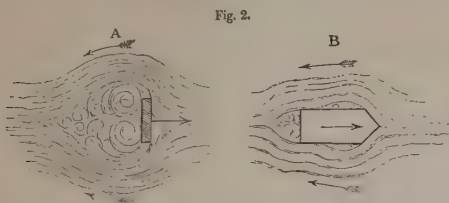


Fig. 2.

block with a wedge-shaped bow having angular shoulders, driven in the same direction.

EXAMPLE A.—The flat plate, A, in the course of each second, impresses on a volume of water bearing a certain proportion to its area and speed of motion, a forward velocity equal to its own velocity; and the particles of water so impelled, while still possessing that forward velocity, glance off from the plate at its edge, and act upon it no more. If we conceive (as stated in Article 152) that the plate is fixed, and that the fluid moves past it in a backward current, the motion of the fluid relatively to the solid will take place in the manner shown in the figure, the current being divided into diverging streams, which re-unite at a certain distance astern of the plate, and are separated from it by a mass of eddies.

In this case the resistance, being equal to the momentum impressed on the fluid in a second, has the following value:—

$$\text{Constant factor} \times \text{Area of Plate} \times \text{Heaviness of fluid} \times \text{Sq. of Velocity} \\ \text{Gravity (32.2)}$$

$$= \text{Constant factor} \times \text{Area} \times \text{Heaviness} \times 2 \times \text{Height due to velocity.}$$

The value of the constant factor is known by experiment to be about '627; that is, the plate impresses a forward velocity equal to its own on a stream of water whose area is '627 of the area of the plate.

EXAMPLE B.—The block, B, being supposed to have a sectional area equal to that of the plate A, its wedge-formed bow impresses, in each second, on a quantity of water nearly the same with that in the preceding example, a velocity whose *forward component* is believed to be about equal to the velocity of the block multiplied by the versed sine of the angle which each side of the wedge makes with the direction of motion. The particles so impelled glance off from the block at its angular shoulders, forming diverging streams which reunite astern of the block; the spaces behind the shoulders and flat stern being filled with eddies.

The resistance in this case is believed to be less than that in the former example nearly in the proportion of the versed sine of the angle of obliquity of the wedge to radius.

The kind of resistance described in this Article never acts upon a well-designed ship; for such a ship is so formed, that the particles of water glide over her surface throughout its whole length, and are left behind her with no more motion than such as is unavoidably impressed upon them through adhesion and stiffness; and hence the failure of the earlier theories of the resistance of ships, which were founded on experiments made with flat plates, wedges, and blocks of various "unfair" shapes.

156. Resistance due to Waves.—Every ship in motion is accompanied by a train of waves, whose production depends on the following general principle:—Let the ship be conceived to be fixed, and the water to flow past it, as stated in Article 152. The velocity assumed by each particle at each instant then depends upon its position relatively to the ship. At each point where the flow of the particles of water is retarded by the action of the vessel upon them, there is an *increase of head*, either in the shape of increased pressure or of elevation of level; and at each point where the flow of the particles of water is accelerated, there is a *diminution of head*, either in the shape of diminished pressure or of depression of level; and those alterations of head are equal to the "heights due" to the alterations of velocity which they accompany.

At the upper surface of the water, where the pressure, being equal to that of the atmosphere, is uniform, the increase or diminution of head consists wholly in elevation and depression of level; and hence, at every place where the velocity of flow of the water past the vessel is a *minimum*, a *wave-crest* is formed; and at every place where the velocity of flow of the water past the vessel is a *maximum*, a *wave-trough* is formed.

A train of waves which simply accompanies the motion of a vessel, without spreading sideways, is not a cause of sensibly increased resistance after it has once been started; for, agreeably to what has been stated in Article 154, long waves, such as are raised by ships, are capable of travelling great distances without sensible loss of energy. But when the waves spread out obliquely from the ship as she advances, the energy employed in raising them is transmitted to distant masses of water, and lost; and fresh energy has to be supplied by the propelling power of the ship to keep up the waves; so that the formation of oblique diverging waves is a cause of permanent resistance.

Every conceivable ship impresses some motion in the direction of her own advance on the particles of water at and near her bow and stern; while the particles amidships move to a greater or less degree backwards; so that the flow of the water, relatively to the vessel considered as fixed, is retarded at the bow and stern, and accelerated amidships. The retardation at the stern is somewhat greater than at the bow, because of the particles at the stern having undergone the action of adhesion.

Hence it follows that every ship is *necessarily* accompanied by two wave-crests and an intervening trough; in other words, a *leading wave* and a *following wave*. These are the waves whose properties were discovered by Mr. Scott Russell, and form the foundation of his "wave-line system" of shipbuilding; the leading wave being a long smooth swell, or "wave of translation," and the following wave a wave of rolling (combined, however, with some translation) of a somewhat steeper and more crested figure.

These (together with some small wrinkles or ripples depending on the superficial cohesion of the water, and called by Mr. Scott Russell "capillary waves") are the only waves which *necessarily* move along with a well-formed ship. But round a ship of a

defective form, there are additional places of minimum and maximum velocity of gliding of the water-particles, produced by short or blunt or convex bows, abrupt changes of direction or of curvature, and other defects in point of fairness of form; and at each of those places there is produced an additional wave-crest or wave-trough, as the case may be; and thus are raised what may be called "*supernumerary waves*."

Figs. 3 and 4 represent a vessel driven at a speed for which her figure and dimensions are ill-adapted, and accompanied by a complex train of waves, whose size is somewhat exaggerated for the sake of distinctness. In the plan, Fig. 3, the leading wave or swell, and the trough which follows it, are hidden under a number of oblique diverging supernumerary waves; but in the elevation, Fig. 4, the form of that swell and trough are indicated by the different levels of the crests of the supernumerary waves. Immediately astern of the vessel is the crest of the following wave, high and steep, and somewhat obliquely diverged, but not so much so as the supernumerary waves. A few additional rolling waves of smaller size follow in the wake of the vessel.

Fig. 3.

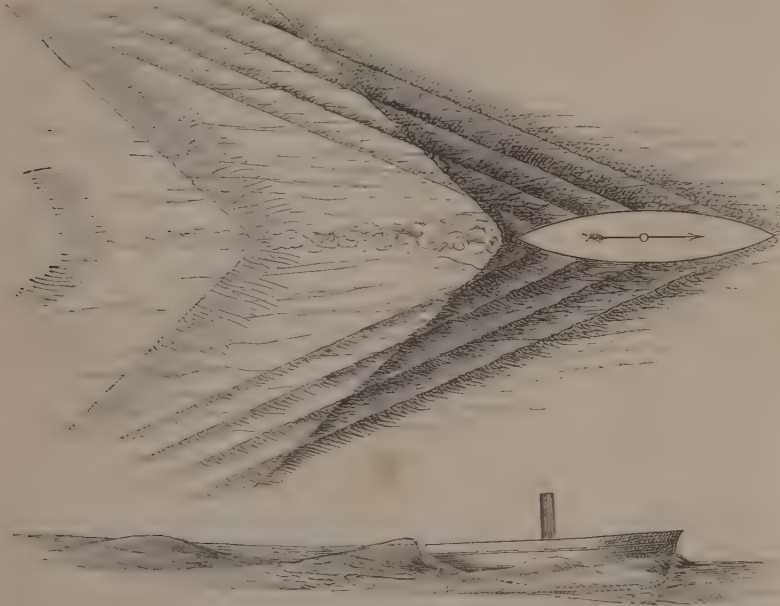
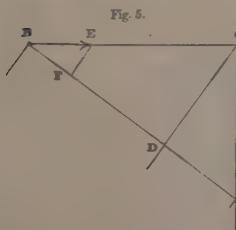


Fig. 4.

Fig. 5 shows the manner in which energy is lost, through the production of oblique diverging waves. Suppose a ship to be moving with a given speed in the direction from B to C; and let C be a point at or near the ship's side, where a wave-trough is produced, and B a point where a wave-crest is produced, by the action of the ship on the water.



Then BC is the half-length, measured parallel to the ship's motion, of a wave which is made to accompany the ship. If BC is equal to, or greater than, the half-length of a natural wave which travels with the same speed, the artificial wave will accompany the ship of itself, without

any sensible increase of resistance, after it has once been started. But if BC is less than the half-length of a natural wave which travels

with the same speed, the artificial wave is *forced* to accompany the vessel, in the following manner:—Draw CL perpendicular to BC; then about B, with a radius, BL, equal to the half-length of the natural wave, which travels with the speed of the vessel, draw an arc of a circle cutting CL in L; join BL, and upon it let fall the perpendicular, CD.

Then—

$$\overline{BL} : \overline{BD} :: \overline{BC}^2 : \overline{BD}^2;$$

but the lengths of natural waves in deep water are as the squares of their velocities; therefore BD is the half-length of a natural wave, whose speed of advance is to that of the ship as BD to BC. The artificial wave, then, keeps pace with the ship by *diverging obliquely* in the direction BL, its trough and crest lying in lines parallel to CD.

The sine of the angle of divergence is—

$$\frac{\overline{CL}}{\overline{BL}} = \frac{\sqrt{(\overline{BL}^2 - \overline{BC}^2)}}{\overline{BL}};$$

The expenditure of energy in producing this divergent wave takes place in the following manner:—The energy of a single rolling wave, affecting a given depth of water, is proportional to its breadth (*along the crest*) and to the square of its height, agreeably to what has been stated in Article 136, Rule XII. Let \overline{BE} represent an unit of length (as one foot), and from E let fall EF perpendicular to BL. Then while the ship advances through the distance \overline{BE} , the energy of the particles of water at B is transmitted to the particles at F; so that all the particles between F and E must receive their energy from the ship, at the expense of the propelling power; in other words, the propelling power has, for each unit of advance of the ship, to raise the breadth, FE, of *new wave*; and that breadth is the sine of the angle of divergence of the wave. Hence the production of an oblique divergent wave causes a resistance to the ship's motion, which is proportional to the square of the height of the wave, the depth of the layer of water affected by it, and the sine of the angle of divergence; the value of which

sine has already been given, as depending on the deficiency of length of the artificial wave, compared with the natural wave, whose speed is equal to that of the ship.

[In algebraical symbols, let h denote the height of a rolling wave, from trough to crest; z , the depth of a layer of water in which the vertical height of disturbance is h ; w , the weight of a cubic foot of water; then the energy possessed by a single wave of the breadth, b , measured along the crest, is—

$$\frac{\pi w b z h^2}{2};$$

and as for each unit of distance advanced by the ship, we have—

$$b = \frac{\sqrt{(\overline{BL}^2 - \overline{BC}^2)}}{\overline{BL}};$$

the resistance caused by the wave in question, so far as it affects the layer of the depth z , is—

$$\frac{\pi w z h^3}{2} \cdot \frac{\sqrt{(BL^2 - BC^2)}}{BL}.$$

The conclusion to be drawn from these principles is, that for each vessel there is a certain limit of speed, below which the resistance due to the production of waves is insensible; and that as soon as that limit is exceeded, that resistance begins to act, and increases at a very rapid rate with the excess of speed. Through the discoveries of Mr. Scott Russell, a vessel can be designed in which this kind of resistance shall be insensible up to a given limit of speed; and therefore the resistance due to waves has no sensible action on a well-formed ship. This will be again considered further on.

157. *The Resistance due to Frictional Eddies* remains alone to be considered. That resistance is a combination of the direct and indirect effects of the adhesion between the skin of the ship and the particles of water which glide over it; which adhesion, together with the stiffness of the water, occasions the production of a vast number of small whirls, or eddies, in the layer of water immediately adjoining the ship's surface. The velocity with which the particles of water whirl in those eddies, bears some fixed proportion to that with which those particles glide over the ship's surface; hence the actual energy of the whirling motion impressed on a given mass of water at the expense of the propelling power of the ship, being proportional to the square of the velocity of the whirling motion, is proportional to the square of the velocity of gliding; in other words, it is proportional to the *height due* to the velocity of gliding. The velocity of gliding of the particles of water over a given portion of the ship's skin, bears a ratio to the speed of the ship depending on her figure, and on the position of the part of her skin in question; and the height due to the velocity of gliding is equal to the height due to the speed of the ship, multiplied by the *square* of the same ratio. Further, the mass of water upon which whirling motion is impressed by a given part of the ship's skin while she advances through an unit of distance, is proportional to the area of that part of the skin, multiplied by the before-mentioned ratio which the velocity of gliding of the water past that part of the skin bears to the velocity of the ship.

Hence the *resistance to the motion of the ship, due to the production of frictional eddies by a given portion of her skin*, is the product of the following factors:—

- I. The area of the portion of the ship's skin in question;
- II. The *cube* of the ratio which the velocity of gliding of the particles of water over that area bears to the speed of the ship; being a quantity depending on the figure of the ship and the position of the part of her skin under consideration;
- III. The height due to the ship's speed; that is,

$$\frac{(\text{speed in feet per second})^2}{64.4}$$

or, $\frac{(\text{speed in knots})^2}{22.6}$;

- IV. The heaviness (or weight of an unit of volume) of the water; (64 lbs. per cubic foot for sea-water);

- V. A factor called the *coefficient of friction*, depending on the material with which the ship's skin is coated, and its condition as to roughness or smoothness.

The sum of the products of the Factors I. and II. for the whole skin of the ship, has of late been called her *AUGMENTED*

SURFACE; and the Eddy-resistance of the whole ship may therefore be expressed as the product of her *Augmented Surface* by the Factors III. IV. and V. above-mentioned.

[In algebraical symbols, let

- $d s$ denote the area of a small portion of the ship's skin;
 q , the ratio which the velocity of gliding of the water over that portion bears to the speed of the ship;
 c , the speed of the ship;
 g , gravity;
 w , the heaviness of the water;
 f , the coefficient of friction; then

$$\text{Eddy-resistance} = f w \frac{c^2}{2g} \int q^3 d s;$$

$\int q^3 d s$ being the *Augmented Surface*.]

The resistance thus determined, being deduced from the work performed in producing eddies, includes in one quantity both the direct adhesive action of the water on the ship's skin, and the indirect action, through increase of the pressure at the bow and diminution of the pressure at the stern.

The existence of this kind of resistance has been recognized from an early period. Beanfoy made experiments on models to determine its amount; Mr. Hawksley and Mr. Phipps have included it in a formula for the resistance of ships; and Mr. Bourne pointed out that it must depend mainly on the ship's immersed girth. But the earlier researches, both experimental and theoretical, throw little light on the subject, and fail to give a trustworthy value of the coefficient of friction; because in them it was assumed that the frictional resistance was proportional to the *actual immersed surface* of the vessel, and the variations of the speed of the gliding of the water over different parts of that surface were neglected.

When the Editor of this treatise (having occasion to compute, in 1857, the probable resistance at a given speed of a steam-vessel built by Mr. J. R. Napier), introduced for the first time the consideration of the *augmented surface*, he adopted, for the coefficient of friction, the constant part of the expression deduced by Professor Weisbach from experiments on the flow of water in iron pipes, viz.,

$$f = .0036;$$

and that value has given results corroborated by practice, for surfaces of clean painted iron. For clean copper sheathing, and for very smooth pitch, it appears probable that the coefficient of friction is somewhat smaller; but there are not sufficient experimental data to decide that question exactly. Experimental data are also wanting to determine the precise increase of the coefficient of friction produced by various kinds and degrees of roughness and foulness of the ship's bottom; but it is certain that that increase is sometimes very great.

The preceding value of the coefficient of friction leads to the following very simple rule for clean painted iron ships:—*At ten knots, the eddy-resistance is one pound avoirdupois per square foot of augmented surface; and varies, for other speeds, as the square of the speed.*

SECTION II.—ADAPTATION OF DIMENSIONS AND FORM TO SPEED.

158. *Wave-Theory as to Dimensions*.—The wave-line system of shipbuilding, as explained by its discoverer, Mr. Scott Russell, at various times, but especially in the Transactions of the Institution of Naval Architects for 1860 and 1861, consists of two principal

parts, one of which relates to dimensions, and the other to shape; the object of both of those parts of the system being, to avoid, or to render insensible, all resistance except that which is unavoidable. The present article relates to the first of those parts, which may be thus summed up:—*The length of the after-body should be one-half of that of a certain rolling wave, which naturally travels with a speed equal to the greatest speed at which the vessel is to be driven; and the length of the fore-body should be equal to the whole length of a wave of translation which naturally travels with the same speed.*

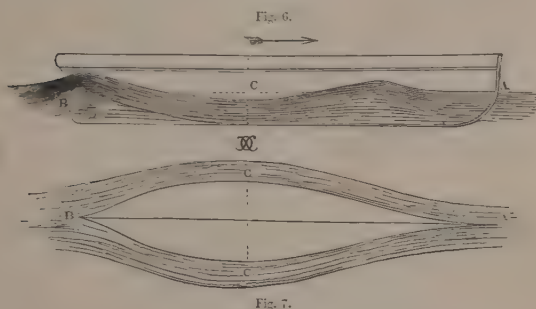
Between the fore-body and the after-body there may be a straight middle-body of any required length.

The principles relating to the after-body and to the fore-body will now be considered separately.

I. AFTER-BODY.—According to the rules and tables given by Mr. Russell for the length of the after-body, that length should be *two-thirds of the circumference of a circle whose radius is twice the height due to the greatest speed of the ship.*

This is the half-length of a rolling wave, which naturally travels, in water of a depth equal to about one-sixth of a wave-length, with a speed equal to the greatest speed of the ship. Hence the result of observing this rule for the length of the after-body is, that in water whose depth is not less than one-third of the length of the vessel's after-body, the *following wave* which the ship raises (as already described in Article 156), is not made to travel faster than its natural speed; and consequently, that there is no waste of energy through the diverging of that wave as explained in the same Article; and also, that the crest of the following wave follows close under the ship's stern, supporting it and pressing it forward, so as to balance the backward pressure of the leading wave.

(In Figs. 6 and 7, the following wave is represented by CB).



To find the length of after-body suited to a given maximum speed: from the Table in Article 136 take *two-thirds of the wave-length* given in the last column of that Table.

It is amply proved by experience, that the observance of the rule of the wave-line system as to the length of the after-body is essential to economy of power; and that the neglect of that rule leads inevitably either to great waste of power in attaining a given speed, or to a great shortening of speed with a given power. There is reason to believe, indeed, that shortness of the after-body is one of the most frequent causes of failures of that sort.

For example, in two very beautifully-shaped and well-executed steamers, which may be designated as X and Y, the expenditure of engine-power proved on trial to be rather more than a fourth part greater than it ought to have been in vessels well adapted to their speed.

	Feet.
X was driven at 15 knots; therefore the length of her after-body should have been $\frac{2}{3} \times 126 =$	84
The actual length of her after-body was	60
Deficiency	24
Y was driven at nearly $17\frac{1}{2}$ knots; therefore the length of her after-body should have been $\frac{2}{3} \times 172 =$	115
The actual length of her after-body was	70
Deficiency	45

A vessel may have an after-body long enough for her speed in water of ordinary depth, but not long enough for the same speed in very shallow water. When such a vessel, at full speed, passes suddenly into very shallow water, the following wave-crest is seen at once to lag somewhat behind the vessel, and to divide into a pair of oblique diverging waves; and the stern of the vessel drops into the trough in front of the following wave-crest, so that she is placed in a position like that of constantly climbing up-hill.

The wave-crest following astern of a vessel that has too short an after-body, may become a cause of danger when the ship is running before the sea; for a long and swift sea-wave, overtaking the vessel, may rise so high in passing over the following wave-crest as to be made to break over her stern.

II. FORE-BODY.—The principle laid down by Mr. Russell is, that the fore-body should have a length equal to that of the *wave of translation*, whose natural speed is equal to the greatest speed of the vessel; that is to say, the circumference of a circle whose radius is twice the height due to that speed, or once-and-a-half the length of the after-body (see the Table in Article 136). The effect of this is to make the bow of the ship extend through the swell which it raises into the still water ahead, so that each particle of water is set in motion in a gradual and continuous manner; and this prevents the formation of the *supernumerary waves* mentioned in Article 156, or at all events of supernumerary waves of such a size as to carry off any appreciable quantity of energy.

When the fore-body is shorter than the length given by this rule, the effect is a certain abruptness or discontinuity in the communication of motion to the particles of water at and near the bow, resulting in the production of at least one oblique diverging supernumerary wave. The resistance due to such a wave increases as the fourth power of the speed.

So far, however, as the Editor of this treatise has been able to gather from the performance of actual steamers, the additional resistance produced by deficiency of length in the fore-body is not nearly so marked or so great as that produced by deficiency of length in the after-body; so long, at all events, as the fore-body is not less than two-thirds of the length given by the rule (that is, equal to the after-body).

For example, a very successful steamer (which will be designated as V), was driven at a speed of

$14\frac{1}{2}$ knots.

	Feet.
The length of the <i>wave of translation</i> which travels with that speed, is	120
And the half-length of the rolling-wave, which, in water whose depth is about one-third of a wave-length travels with the same speed, is about	80
This latter length was the actual length of the after-body, which consequently was in exact accordance with the rule.	
The fore-body, however, instead of being 120 feet long, was of the same length with the after-body, or	80
The vessel had no middle-body.	

The engine-power required to drive this vessel was only about 5 or 6 per cent. greater than that estimated from eddy-resistance alone.

It would seem, then, that in practice, while it is essential to economy of power that the rule of the wave-system as to the least length of the after-body should be inflexibly observed, the rule as to the length of the fore-body admits of some latitude; and although it may be best, with a view to saving of resistance alone, to make the fore-body of the full length of the wave of translation, still, if there be any reason against making the ship so long as that rule would require, the fore-body may be made shorter to an extent not exceeding one-third, care being taken not to increase the greatest obliquity of the water-lines to a fore-and-aft line.

159. *Wave System as to Figure.*—The second part of the wave-line system prescribes for the lines of motion of the particles of water as they glide along the surface of the ship, figures imitated from those of waves; the lines of the after-body being trochoids, like the outlines of rolling waves (already described in Article 135), and those of the fore-body curves of versed-sines; such having been found by experiment to approximate very nearly to the outlines of waves of translation.

As it is known by observation that the particles of water slide past the fore-body of a ship in layers which (at all events near the bow) are approximately horizontal, and past the after-body in converging layers which are nearly vertical amidships, the curve of versed-sines is used for the horizontal sections or water-lines of the fore-body; and the trochoid is used both for the horizontal sections and the vertical sections, or buttock-lines, of the after-body.

As those two kinds of *wave-lines* are curves along which it is known with certainty by observation and experiment that the particles of water are capable of gliding smoothly, their use insures the absence of supernumerary diverging waves, and of the consequent increase of resistance. Methods of constructing those curves will be explained in Division Second.

160. *Varieties of Fair Water-lines—Lissoneoids.*—The strict observance of this second part of the wave-line system appears to be not so important to economy of power as that of the first part; for although it is unquestionable that wave-line curves possess the property of permitting the particles of water to glide smoothly past them, and that of not raising supernumerary waves intermediate between the bow and stern, they are not the only curves which possess those properties.

The figures which permit the smooth gliding of water past them comprehend all those which fulfil a certain differential equation, and are infinite in variety. The mathematical properties of a very numerous class of those figures, presenting approximations to all the forms of ships' water-lines which are usual in practice, have been treated of by the Editor of this work in two papers, one of which was read to the Royal Society in 1863, and the other is now nearly complete.

By the further condition, of not raising intermediate supernumerary waves, the proportions of those figures are to a certain extent limited; but even as so limited, they still present great variety.

The most important of those water-line curves, in a practical point of view, are those which have the *fullest form* (in other words, the greatest coefficient of fineness) *consistent with not raising intermediate supernumerary waves*; curves which the Editor has proposed to call *Lissoneoids*.^a

The coefficients of fineness of those lissoneoids which are of proportions available for practical use (the length being four times the breadth and upwards) lie between $\frac{2}{3}$ and $\frac{3}{4}$, and differ little in any case from '637, the coefficient of fineness of a curve of sines.

Such may be regarded as the higher limits of the coefficient of fineness of water-lines, well formed for economy of power at high speeds, the lower limit being

·5,

which is the coefficient of fineness of the curve of versed-sines used by Mr. Scott Russell for the fore-body.

Methods of constructing the lines referred to in this Article, either exactly or to an approximation near enough for practical purposes, will be explained in the Second Division of the Treatise.

The actual paths of the particles of water in gliding over the bottom of a vessel, are neither horizontal water-lines, nor vertical buttock-lines, but are intermediate in position between those lines, and approximate in well-shaped vessels to geodetic lines—that is, lines of shortest distance, such as are followed by an originally straight strake of plank when bent to fit the shape of the vessel. If, however, the water-lines and the buttock-lines are fair curves, so also must be the geodetic lines. The construction of the latter class of lines also will be explained in the Second Division.

SECTION III.—COMPUTATION OF PROPELLING POWER AND SPEED.

161. *General Explanations.*—The method of calculation now to be explained and illustrated was first practically used in 1857, under the circumstances stated in Article 157. A very condensed account of it, illustrated by a table of examples, was read to the British Association in September, 1861, and printed in various mechanical journals for October of that year; and some further explanations appeared in a paper on Waves in the Philosophical Transactions for 1862.†

The method proceeds by deducing the eddy-resistance (already explained in Art. 157) from an approximate value of the augmented surface. It is therefore applicable to those vessels only in which eddy-resistance forms the whole of the appreciable resistance; but such is the case with all vessels of proportions and figures well adapted to their speed, as has been explained in the preceding sections; and as for misshapen and ill-proportioned vessels, there does not exist any theory capable of giving their resistance by previous computation.

162. *Computation of Augmented Surface.*—To compute the exact augmented surface of a vessel of any ordinary shape, would be a problem of impracticable labour and complexity. The method employed, therefore, as an approximation for practical purposes, is to choose in the first instance a figure approximating to the actual figure, but of such a kind that its augmented surface can be calculated by a simple and easy process, and to use that augmented surface instead of the exact augmented surface of the ship; care being taken to ascertain by comparison with experiments on ships of various sizes and forms, whether the approximation so obtained is sufficiently accurate.

The figure chosen for that purpose is the trochoid, or rolling-wave-curve, extending between a pair of crests, such as A and B,

^a From words denoting "smooth-gliding" and "ship-shape."

† A prediction of the speed of the *Great Eastern*, with different amounts of engine-power, obtained by this method of calculation, was published in the *Philosophical Magazine* for April, 1869.

Half-Girths from Body-plan. Feet.	Simpson's Multipliers.	Products.
21-0	1	21-0
27-2	4	108-8
30-8	2	61-6
34-6	4	138-4
38-8	2	77-6
41-5	4	166-0
42-6	2	85-2
44-0	4	176-0
44-0	2	88-0
44-0	4	176-0
43-3	2	86-6
42-1	4	168-4
40-3	2	80-6
38-1	4	152-4
36-0	2	72-0
35-0	4	140-0
32-0	1	32-0
Divide by.....		8) 1830-6 Sum.
Divide by $\frac{1}{3}$ number of Intervals.....		8) 610-2
Mean Immersed Girth.....		76-3
× Length.....		380
Product.....		28994
× Coefficient of Augmentation.....		1-275
Augmented Surface.....		36979 Square Feet.
Indicated Horse-power on Trial.....		5471
× Coefficient of Propulsion.....		20000
Divide by Aug. Surface.....		36979) 109,420,000 Product.
Cube of Probable Speed.....		2959
Probable Speed, Computed.....		14-356 Knots.
Actual Speed, on Trial.....		14-354
Difference.....		-002

EXAMPLE II.—H.M.S. *Fairy* will next be taken as an example, on account of the great contrast in size between her and the *Warrior*.

Displacement.....	168 tons.		
Draught of Water.....	4-83 feet.		
Water-lines.	Sine of Obliquity,	Square of Sine.	Fourth Power of Sine.
L.W.L 	23 	0529 	0028
2 W.L 	22 	0484 	0023
3 W.L 	21 	0441 	0019
4 W.L 	17 	0289 	0008
Keel 	0 	0 	0
	Means,.....	0304	0015
1 + (4 × 0304) + 0015 = 1.123, Coefficient of Augmentation.			
Length on Water-line,.....	144 Feet.		
× Mean Immersed Girth (measured mechanically with an instrument),.....	19		
× Coefficient of Augmentation,.....	1.123		
Augmented Surface,.....	3072 Square Feet.		
Indicated Horse-power, on Trial,.....	364		
× Coefficient of Propulsion,.....	20000		
÷ Augmented Surface,.....	3072) 7,280,000 Product.		
Cube of Probable Speed,.....	2370		
Probable Speed, computed.....	13.333 Knots.		
Actual Speed, on Trial,.....	13.324		
Difference,.....	009		

The *Fairy* occurs in the table of examples given in the paper of 1861, already referred to: in the present paper the measurements have been revised and improved in precision, especially as regards the coefficient of augmentation. The difference in the result is but small.

EXAMPLE III.—H.M.S. *Victoria and Albert*,—a wooden vessel, sheathed with copper, will now be employed, not to illustrate the computation of probable power at a given speed, or of probable speed at a given power; but to compute a value of the coefficient of propulsion for a copper-sheathed vessel.

Displacement on Trial Trip,.....	1980 tons.		
Draught of Water,.....	{ Forward, 13.8 feet. Aft,..... 14.0 "		
Water-lines.	Sine of Obliquity.	Square of Sine.	Fourth Power of Sine.
L.W.L	19	0361	0013
2 W.L	185	0342	0012
3 W.L	17	0289	0008
4 W.L	14	0196	0004
Keel	0	0	0
Means.....		0252	0008

$1 + (4 \times 0252) + 0008 = 1-102$, Coefficient of Augmentation.

Length on Water-line..... 300 Feet.

× Mean Immersed Girth (measured mechanically
with an instrument)..... 40

× Coefficient of Augmentation..... 1-102

Augmented Surface..... 13224 Square Feet.

× Cube of Speed in Knots..... $17^3 = 4913$

÷ Indicated Horse-power on Trial..... 2980) 64,969,512 Product.

Coefficient of Propulsion..... 21,802

Had the probable speed been computed with the coefficient of propulsion 20000, the result would have been 16-53 knots, instead of 17.

168. *The Resistance of the Air* to the motion of the hull of a vessel, although it forms a considerable part of the whole resistance, is not known with any approach to precision; and it must be held; in the present state of our knowledge, to be comprehended amongst the various additional resistances which are provided for in Article 163, by multiplying the net engine-power required to overcome the resistance of the water by 1-63, or some such coefficient.

If the figure and dimensions of the vessel were the same above and below water, and the surface in the same condition, the resistance of the air might be estimated to be less than that of the water in the ratio in which the heaviness of air is less than that of water—viz., about $\frac{1}{800}$ th; but the resistance of the air must always be greater than that fraction of the resistance of the water, because no vessel is so favourably shaped for gliding through a fluid above water as she is below water.

When a large flat surface (such, for example, as the forward end of a deck-house) is driven through the air in a direction perpendicular to itself, the resistance may be estimated approximately by multiplying together the area of the surface, the height due to its velocity ($\frac{\text{Velocity in knots}^3}{22-6}$), the heaviness of air (say 075 lbs. per cubic foot), and the coefficient 1-25. Such surfaces, however, ought not to be presented to the air by vessels intended for high speed. In still air, the velocity above referred to is the same with that of the ship; in wind, it is the direct component of the apparent velocity of the wind relatively to the ship, of which more will be said further on.

169. *The Resistance in Rough Water* is increased beyond the resistance in smooth water by two causes—the revolutions of the particles of water, and the alteration of the figure of the immersed surface of the ship, through her own oscillations and those of the waves.

I. *The Revolution of the Particles of Water* tends to increase the resistance of the ship in the proportion which the sum of the

squares of the velocities of the ship and of the particles of water in the "effective wave-surface" (Article 140) bears to the square of the velocity of the ship. For example, suppose a vessel to be moving at a speed of 10 knots amongst waves whose period is 10 seconds and height 16 feet. The velocity of the particles of water is 3 knots; and the resistance of the vessel will probably be increased nearly in the proportion—

$$\frac{10^2 + 3^2}{10^2} = 1.09 : 1.$$

II. The rolling and pitching of the ship, her permanent heeling under the pressure of the wind, and the motions of the waves, cause different portions of the ship's surface to be immersed from those which are immersed in smooth water and calm weather; and thus the resistance may be considerably increased, unless care has been taken in designing the ship to make not only the lines which are always under water, but all the lines which at any time may be under water, of proper dimensions, fair figures, and sufficient fineness. Thus, flaring bows, and a bluff, overhanging stern, if carried to excess, although they may have little or no bad effect in smooth water and fair weather, may cause a serious falling off in speed during stormy weather.

170. *Proportions of Length to Breadth.*—Principles which have been explained in Article 158 fix the *least absolute length* suitable for a vessel which is to be driven at a given speed. But after that least length has been fixed, a question may arise as to whether that least length, or a greater length, is the most economical of power. That question is answered by finding the proportion of length to breadth, which gives the *least augmented surface* with the required displacement.

That proportion can be found in an approximate way only; because of the approximate nature of the process by which the augmented surface itself is found. The following are some of the results obtained in certain cases:—

I. When the proportion of breadth to draught of water, and the figure of cross section, are fixed, so that the mean girth bears a fixed proportion to the breadth, it appears that the proportion of length to breadth which gives the least augmented surface for a given displacement, is about 7 to 1.

II. When the absolute draught of water is fixed, the proportion of length to breadth which gives the least augmented surface for a given displacement depends on the proportion borne by the draught of water to a mean proportional between the length and breadth, and on the figures of the cross sections. The following are some examples for flat-bottomed vessels:—

$\sqrt{\frac{\text{Length} \times \text{Breadth}}{\text{Draught}}}$	from 4 to 5;	7 to 10;	12 to 16;	17 to 23;
$\frac{\text{Length}}{\text{Breadth}}$	7;	8;	9;	10.

III. By cutting a vessel in two amidships, and inserting a straight middle body, the proportion borne by her resistance to her displacement is always diminished; because the midship section has a less mean girth in proportion to its area than any other cross section of the ship; and therefore the new middle body adds proportionally less to the augmented surface than it does to the displacement.

IV. It does not follow, however, that a straight middle between tapering ends is the most economical form; for by adopting continuous curves from bow to stern for the water-lines, instead of the lines compounded of curved ends and a straight middle, the

same length, the same displacement, and almost exactly the same mean girth may be preserved, and the obliquity of the water-lines at the entrance diminished.

171. *Cross Sections of Least Girth.*—When the principal object in designing a ship is to diminish the resistance as much as possible, it becomes useful to know the proportions of a cross section which give it the least immersed girth in comparison with its immersed area. It is well known that when the form to be adopted for the cross section is not limited by any other conditions besides that of having the least immersed girth possible for its area, the figure which fulfils that condition is a semicircle. A complete semicircle, however, is not an eligible form in other respects, being subject to the defects either of crankness or of excessive rolling, one of which cannot be cured without producing the other.

One of the most common forms of cross section for merchant ships is that shown in Fig. 4b of Chapter III., Article 97, page 39, being a rectangle with rounded corners at the bilges. When the proportion of breadth to draught in such a rectangle is fixed, Mr. James R. Napier has shown* that the radius of bilge which makes the least girth for a given area is given by the following formula:—

[Let B denote the breadth on the water-line;

H, the draught of water;

R, the radius of bilge; then

$$R = \frac{B + 2H - \sqrt{(B + 2H)^2 - 1.7168 BH}}{0.8584}.]$$

The following are examples:—

B ÷ H,	$\frac{1}{4}$	$\frac{1}{2}$	1	2	3	4	unlimited;
R ÷ H,	.114	.23	.35	.54	.63	.70	1.0.

When B ÷ H diminishes without limit, R approximates to the limit $\frac{1}{2}$ B.

172. *Trials of Speed—Effect of Tidal Currents.*—The present Article relates to the accurate determination of speed alone. The measurement of propelling power, and the comparison between speed and power, will be treated of further on.

Trials of speed are usually made in calm or moderate weather, by running a certain measured distance in a straight line; the two ends of the run being marked by fixed objects, such as lighthouses, beacons, or buoys. The distance run in nautical miles and fractions of a mile being divided by the time of running it in hours and fractions of an hour, gives the mean speed in knots of a single run. The result of every single run is affected by various errors; and in order to eliminate those errors as far as possible, it is necessary to make several runs, and combine their results in such a way as to make the errors counteract each other.

In order that the errors produced by tidal currents may be capable of being eliminated by a simple process, the distance run should lie *along* their direction, and not *across* it, nor *obliquely*; and it should therefore be situated in a channel where the tidal currents run alternately in one direction and in the opposite direction only, and do not at any time flow transversely or obliquely. *The tideway should also be one in which the tidal currents move according to the ordinary laws, and do not present the irregularities that are sometimes produced by disturbing causes.

When those conditions are fulfilled, although the times and velocities of the tide should be unknown, the speed deduced from

* Proceedings of the Philosophical Society of Glasgow, December, 1862.

any *three* runs, two in one direction and one in the opposite, may be so combined as to eliminate the action of the tidal current, and give a mean speed which is affected by other causes of error only, such as the force of the wind, irregularities in the engine-power, inaccuracies of observation, &c. To eliminate, as far as possible, the effects of those causes of error also, four or more runs must be made, and combined in sets of three; from each set of three there is to be deduced a velocity freed from tidal influence; and the mean of those velocities will be the most probable value of the true speed.

When the *time of slack water* is known, but not the velocity of the tidal current, the effects of the tide can be eliminated by combining the results of two runs. When the time of slack water and the greatest velocity of the current are both known, the effects of the tide can be calculated and allowed for in each single run.

Four kinds of cases may be distinguished—those in which the distance run is short, such as a single nautical mile, so that several runs can be made over it at short and nearly equal intervals of time apart; those in which the distance run is considerable, such as twelve or thirteen nautical miles, the time and velocity of the tide being unknown; those in which the time of slack water is known, but not the velocity of the tide; and those in which both those circumstances are known.

CASE I.—When the runs are made over a short distance, such as a “measured mile,” and at nearly equal intervals of time, take the mean of the speeds of the first and second runs, the mean of the speeds of the second and third runs, of the third and fourth runs, and so on: then take the means of those means, or *second means*. Those second means, which are free from tidal influence, are to be added together and divided by their number for the final mean.

CASE II.—But when the distance run is great, such as the 13·66 nautical miles between the Cloch Lighthouse and the Cumbrae Lighthouse in the Firth of Clyde, the second means are not sufficiently accurate, especially if the intervals of time between the runs are to any considerable extent unequal. In such cases the correct rule is as follows:—

- (1.) Calculate the apparent speed of each of the runs as usual, by dividing the distance by the time; and group them in sets of three. For example, 1, 2, 3; 2, 3, 4; 3, 4, 5; &c.
- (2.) Each set of three is to be treated as follows:—Find the two intervals of time between the middle instants of the first and second, and of the second and third runs of the set. Reduce those intervals to the corresponding angular intervals by the following proportion:—

As $12^h\ 24^m$ (the duration of a tide)

: is to a given interval of time

: : so is 360°

: to the corresponding angular interval.*

- (3.) Multiply the *first* apparent speed by the cosecant of the first angular interval;

The *second* apparent speed by the sum of the cotangents of the two angular intervals;

The *third* apparent speed by the cosecant of the second angular interval.

- (4.) Add together the products, and divide their sum by the sum of the before-mentioned multipliers; the quotient will be a speed from which tidal effects have been eliminated.

- (5.) Add together the velocities deduced from the sets of three runs, and divide by their number, for a final mean.

Cases may occur in which intervals of more than a quarter tide, or $3^h\ 6^m$, elapse between the middle instants of two runs of a set. In such cases it becomes necessary to subtract, instead of adding, certain multipliers and products—in algebraical language, certain multipliers and products in the calculation become negative instead of positive. The principles according to which it is to be decided whether a given multiplier is positive or negative are those of ordinary trigonometry, and are summed up in the following table:—

Time.		Angles.		Cosecants.	Cotangents.
Between	$0^h\ 0^m$	{	Between 0°	positive.
and	$3\ 6$	}	and 90	positive.
Between	$3\ 6$	{	Between 90	positive.
and	$6\ 12$	}	and 180	negative.
Between	$6\ 12$	{	Between 180	negative.
and	$9\ 18$	}	and 270	positive.
Between	$9\ 18$	{	Between 270	negative.
and	$12\ 24$	}	and 360	negative.

CASE III.—When the *time of slack water* is known, but not the velocity of the current, the effect of the tide can be eliminated from two runs, by proceeding as follows:—

- (1.) Calculate the intervals of time from slack water to the middle instants of the two runs respectively, and reduce them to the corresponding angles, as in Case II.
- (2.) Multiply the apparent speed of each run by the *sine* of the angular interval belonging to the *other* run.
- (3.) If the runs were both made in the same current (ebb or flood), divide the sum of the products by the sum of the multipliers; if the tide turned between the runs, divide the difference of the products by the difference of the multipliers; the quotient will be a speed from which tidal action has been eliminated.
- (4.) When three or more runs have been made, they are to be combined by pairs, the result of each pair computed by the preceding rules, and the arithmetical mean of the results taken as the probable true speed.

CASE IV.—When all particulars as to the tidal currents are known, velocities as well as times, the speed of a ship through the water can be deduced from one run, by computing and allowing for the distance that she is drifted by the tide during the run, and dividing the corrected distance run by the time. When the data as to the tide are, the time of slack water and the greatest velocity of the current, the following is the rule for computing the distance that the vessel is drifted during a given run:—

Calculate the intervals of time elapsed from slack water to the beginning and end of the run respectively; convert those intervals into the corresponding angles, as in Case II.; multiply the greatest speed of the current by the difference of the cosines of those angles (if both are acute or both obtuse); or by the sum of those cosines, if one angle is acute and the other obtuse; the product will be the distance drifted.†

The speed of a vessel through the water may be measured directly by means of the log.

* The following rule is convenient:—Divide the time in seconds by 124; the quotient will be the angle in degrees.

† The demonstration of the rules for Cases II., III., and IV. is given in a paper by the Editor of this treatise, published in the Transactions of the Institution of Engineers in Scotland for 1864.

It may also be measured by means of a modification of "Pitot's Tube," first applied to this purpose by Mr. J. R. Napier. That instrument consists mainly of a mercurial pressure-gauge, in which the height of a column of mercury indicates the pressure exerted by the water in a tube whose lower end passes through the bottom of the vessel, and turns horizontally, with an open mouth facing horizontally ahead. The velocity is *nearly* proportional to the square root of the pressure indicated; but the best way to determine the relation between the velocity and the pressure, is to observe the height of the mercurial column during some runs made at different speeds along a measured mile. When two or three points on the scale have thus been determined, the intermediate divisions can easily be laid down. The instrument, therefore, does not serve to ascertain the speed of a vessel precisely during a first trial trip; but having been graduated by the aid of observations made during such a trip, it furnishes a convenient means of observing the speed at any future time.

SECTION IV.—PROPULSION BY THE REACTION OF THE WATER.

172. *Reaction of the Water—Slip.*—The present Section relates to those properties alone which all propellers have in common; the peculiarities of special propellers belong to the Sixth Division of the Treatise.

In Chapter I., Article 9, it has already been stated that all propellers act by driving water astern, and that it is the reaction of the water so driven back, which, being transmitted through the propeller to the framework of the machinery, drives the vessel ahead. When the vessel, therefore, is moving at an uniform speed, the reaction of the water is equal to her resistance.

The velocity, *relatively to still water*, of the backward current of water produced by the propeller, is called the *slip*.

The velocity of the backward current *relatively to the slip*, is the sum of the slip and the ship's speed; and this is also called the *velocity of the propeller*.

The amount of the reaction of the water is equal to the backward momentum impressed upon it in a second; and that momentum is the product of two factors—the weight of water driven backward in a second, and the ratio which the slip bears to the accelerating effect of gravity in a second. In determining the weight of water driven backward in a second, two cases may be distinguished, according as the propeller lays hold of sensibly still water, or of water which is in motion either with or against the ship.

CASE. I.—Still Water.—In this case the weight of water driven backwards in a second is the product of the speed of the ship in feet per second, the sectional area of the stream of water upon which the propeller acts in square feet, and the heaviness of the water in lbs. to the cubic foot. Hence the reaction of the water has the following amount:—

$$\frac{\text{Heaviness} \times \text{Area of stream} \times \text{Speed} \times \text{Slip (both in feet per sec.)}}{\text{Gravity (32.2 feet per second)}}$$

The resistance of the ship, to which that reaction is equal, has the following value, according to Articles 157 and 163:—

$$\frac{\text{Coefficient of friction} \times \text{Heaviness} \times \text{Augmented surface} \times \text{Speed}^2}{\text{Gravity} \times 2}$$

Striking out the common factors from those two equal quantities we obtain the following equation:—

$$2 \times \text{Area of stream} \times \text{Slip} = \text{Coeff. of friction} \times \text{Augmented surface} \times \text{Speed of Ship;}$$

which may be otherwise expressed thus, in the form of a proportion—

$$\begin{aligned} &\text{As twice the sectional area of the reacting stream} \\ &: \text{is to the augmented surface of the ship multiplied by} \\ &\quad \text{the coefficient of friction (.0036 nearly for clean iron} \\ &\quad \text{ships)} \\ &: : \text{so is the speed of the ship} \\ &: \text{to the slip of the propeller.} \end{aligned}$$

[In algebraical symbols, let c denote the speed of the ship, s the slip; f the coefficient of friction, a the area of the reacting stream, A the augmented surface of the ship; then

$$2as = fAc; \text{ and } 2a : fA :: c : s.]$$

Hence it appears, that in order that the slip of the propeller may bear a given proportion to the speed of the vessel, the area of the reacting stream must bear *half the reciprocal* of that proportion to the augmented surface multiplied by the coefficient of friction.

[In algebraical symbols

$$a = \frac{fAc}{2s}.]$$

EXAMPLES, taking the coefficient of friction = .0036.

Augmented Surface ÷ Sectional Area of Reacting Stream.	Slip ÷ Speed of Ship.	Slip ÷ Speed of Propeller.
240	0.432	0.302
220	0.396	0.269
200	0.360	0.265
180	0.324	0.245
160	0.288	0.224
140	0.252	0.201
120	0.216	0.177
100	0.180	0.153
80	0.144	0.126
60	0.108	0.097

CASE II.—When the propeller lays hold of water that has been already *set in motion by the ship*, the following modifications are to be made in the expression for the *reaction of the water*; for a speed equal to that of the ship, is to be substituted the original speed of the particles of water relatively to the ship; and for the *apparent slip*, or slip relatively to still water, is to be substituted the *real slip*, or total change produced by the propeller in the velocity of the particles of water. Thus is obtained the following equation:—

$$2 \times \text{Area of stream} \times \text{Relative speed of stream} \times \text{Real slip} \\ = \text{Coeff. of friction} \times \text{Aug. surface} \times (\text{Speed of ship})^2.$$

[In algebraical symbols, let u be the velocity impressed on the particles of water before the propeller lays hold of them; then, using the $\left\{ \begin{smallmatrix} \text{upper} \\ \text{lower} \end{smallmatrix} \right\}$ signs according as the motion denoted by u is $\left\{ \begin{smallmatrix} \text{ahead} \\ \text{astern} \end{smallmatrix} \right\}$, we have—

$$\begin{aligned} &\text{Relative speed of stream, } c \mp u; \\ &\text{Real slip, } s \pm u; \end{aligned}$$

and consequently

$$\begin{aligned} 2a(c \mp u)(s \pm u) &= fAc^2; \\ a &= \frac{fAc^2}{2(c \mp u)(s \pm u)}; \\ s &= \frac{fAc^2}{2a(c \mp u)} \mp u. \end{aligned}$$

It is evident that the previous motion of the particles of water $\left\{ \begin{smallmatrix} \text{ahead} \\ \text{astern} \end{smallmatrix} \right\}$ tends to $\left\{ \begin{smallmatrix} \text{increase} \\ \text{diminish} \end{smallmatrix} \right\}$ the real slip, and to $\left\{ \begin{smallmatrix} \text{diminish} \\ \text{increase} \end{smallmatrix} \right\}$ the apparent slip. Paddle-wheels usually work in water having a slight previous motion astern; the screw, in water which has

a previous motion ahead: and the practical results are in accordance with the principles just stated; the apparent slip of the paddle wheel being in general somewhat greater than that of the screw.

173. *Actual Area and Effective Area.*—The actual area of a propeller, whether paddle or screw, is measured on a plane perpendicular to the direction of motion of the vessel, and is the area contained within the outline of the paddle or screw, as projected on that plane.

The *effective area* is the sectional area, already mentioned, of the stream of water laid hold of by the propeller, and is generally, if not always, *greater than the actual area*, in a ratio which in good ordinary examples is 1·2 or thereabouts, and is sometimes as high as 1·4; a fact probably due to the stiffness of the water, which communicates motion laterally amongst its particles.

The following example is taken from H.M.S. *Warrior*—

$$\begin{aligned} \frac{\text{Slip of screw}}{\text{Speed of ship}} &= 0\cdot12 \\ \text{Augmented Surface,} & \dots\dots\dots 56979 \text{ Square feet.} \\ \times \text{Coefficient of Friction,} & \dots\dots\dots \cdot0036 \\ \hline & \div 2 \times 12 = 0\cdot24 \quad 133\cdot1244 \text{ Product.} \\ \text{Probable Area of Stream taken hold of by Screw,} & \dots\dots\dots 556 \text{ Square feet.} \\ \text{Actual Area of Screw-disc,} & \dots\dots\dots 471 \text{ " "} \end{aligned}$$

$$\frac{\text{Effective area}}{\text{Actual area}} = \frac{556}{471} = 1\cdot18.$$

The ratio of the effective to the actual area depends to a great extent on the figure and construction of the particular propeller used, and will therefore be further discussed in the Sixth Division of this Treatise.

175. *The Efficiency of a Propeller* is the proportion which the useful work performed by it in driving the vessel bears to the whole energy expended in moving it. In the absence of all friction of bearings, and of all action on the water except that of driving its particles directly astern, the efficiency of the propeller would be simply the ratio of the speed of the vessel to the speed of the propeller; that is to say,

$$\frac{\text{Speed of vessel}}{\text{Speed of vessel + Slip}};$$

but in every case there are resistances overcome and transverse motions impressed on the water, which reduce the efficiency to a smaller fraction. This subject will be further considered in the Sixth Division.

The efficiency of a propeller may be very much impaired, if it is so placed, and the vessel so shaped, that the action of the propeller increases the resistance to the motion of the vessel. Such is the case when the lines of the after-body of a screw-steamer are too full; for the action of the screw then causes a diminution of the pressure of the water against the stern of the vessel, which has the same effect with an increased pressure against the bow.

By the *Efficiency of the Engine and Propeller* is meant, the ratio which the useful work of propelling the vessel bears to the whole energy exerted by the steam on the piston, as shown by the Indicator-Diagram. This too will be considered further in the Sixth Division.

176. *Comparative Performance of Steam-ships.*—In ships of similar figures, the augmented surfaces are proportional to the squares of the cube roots of the displacements; and so also are the resistances at a given velocity, so long as the lengths of the vessels do not fall short of the lowest limit suited to that

velocity. With the same limitation, the engine-power increases as the cube of the speed.

If, therefore, the square of the cube root of the displacement be multiplied by the cube of the speed, and divided by the indicated engine-power, a quotient is obtained, whose magnitude is a test of the comparative economy of power in different vessels, as the result of the whole combination of ship, propeller, and engine. That quotient is called the *coefficient of performance*. When the displacement is expressed in tons, the speed in knots, and the power in indicated horse-power, it ranges from 200 to 260, or thereabouts, in good examples. When it falls much short of 200, there is some fault in the ship, the propeller, or the engine.^{*}

The following are some examples, from vessels already cited:—

Ship	Displacement. Tons.	Speed. Knots.	Indicated Horse Power.	Coefficient of Performance.
Warrior,.....	8997 ...	14·354 ...	5471 ...	234
Fairy,.....	163 ...	13·324 ...	864 ...	198
Victoria and Albert,...	1980 ...	17·0 ...	2980 ...	260
V,.....	140 ...	14·5 ...	412 ...	204
X,.....	— ...	— ...	— ...	151
Y,.....	— ...	— ...	— ...	143

SECTION V.—PROPULSION BY THE WIND.

177. *General Explanations.*—In describing the directions from which the wind blows, and the directions towards which a ship's head points, and towards which she moves, and in comparing those directions with each other, angles are often expressed in *points*, and in half and quarter points; a point being *one-eighth of a right angle*, or $11\frac{1}{4}$ degrees. The angle which a ship's course makes with the direction from which the wind blows, is described as being so many points "near the wind."

The propulsion of a vessel by means of the wind takes place through the action of sails, which intercept and turn aside certain streams of air, and are pressed upon by that air with a force depending on the change of momentum undergone by it in a second. The pressure so exerted on the sails is communicated to the vessel; it acts in a direction nearly perpendicular to the surfaces of the sails, and drives the ship in such a direction, and at such a speed, that the resistance of the water is equal and opposite to the driving effort of the wind. That effort may be resolved into two components—a longitudinal component, tending to drive the ship directly ahead; and a transverse component, tending to drive her sideways to leeward; and those components are respectively equal and opposite to the longitudinal and transverse components of the resistance of the water. The shape of the ship is such, that the transverse component of the resistance corresponding to a given velocity is very much greater than the longitudinal component; consequently the "*leeway*," or transverse component of the velocity impressed on the ship is much less than the "*headway*," or longitudinal component of that velocity (being seldom more than *one-tenth*, and rarely reaching *one-fifth*); and the course of the ship, or direction in which she moves, makes a small angle to leeward of the direction of her keel, called the "*angle of leeway*" (seldom much more than half a point, and rarely reaching a point). There is no leeway when a ship runs right before the wind.

The transverse component of the pressure of the wind, and the equal and opposite transverse component of the resistance of the water, form (as already stated in Art. 129) a couple,

* See Atherton On Steam-ship Capability.

which heels the ship over until her moment of stability becomes equal to it and balances it; and the ship is thus made to lean over to leeward, when steady, at a certain angle depending on her stability and on the moment of the heeling couple; and to oscillate, when rolling, about that position instead of about a vertical position. The *power of a ship to carry sail* thus depends on her transverse stability, and on the greatest angle of steady heel consistent with safety and convenience.

The longitudinal component of the pressure of the wind, and the equal and opposite longitudinal component of the resistance of the water, form a couple tending to depress the bow; its effect is comparatively small, because of the moment of the vertical component of the resistance, which tends to elevate the bow, and because of the comparatively great longitudinal stability of the vessel.

Unless the resultant pressure of the wind and the resultant resistance of the water act in the same vertical plane, they form a couple tending to turn the vessel horizontally. When the pressure of the wind acts abaft the resistance of the water, that couple tends to make the ship *come up* to the wind, and she is said to "*gripe*" or to be "*ardent*," when the pressure of the wind acts before the resistance of the water, the couple tends to make the ship *fall off* from the wind, and she is said to be "*slack*." Those tendencies may be counteracted either by means of the rudder, whose action on the water has the effect of modifying the position of the resultant resistance, at the expense, however, of increasing the total resistance and diminishing the speed; or by altering the trim of the ship, for the same purpose; or by suitable trimming of the sails, so as to modify the position of the resultant pressure of the wind. But those processes involve many questions which are foreign to this work, and belong not to Naval Architecture, but to Seamanship. The business of the naval architect is to design a vessel in which they shall be as little needed as possible, and in which, if there is any slight deviation from the total absence of both ardency and slackness, it shall be on the side of ardency; that being a quality which is in some respects useful.

The present section relates to those principles only which are common to all modes of propulsion by sails; matters of detail being reserved for the Fifth Division.

178. *Direct Impulse and Classification of Winds.*—The law which connects the velocity of the wind with its direct impulse against a flat surface at right angles to its motion, is still but imperfectly known. According to the experiments most generally relied on, it may be estimated at about 1·86 *times the weight of a column of air, whose base is equal in area to the surface struck by the wind, and its height that due to the velocity of the wind*; to find which height, the square of the velocity in feet per second is to be divided by 64·4; or the square of the velocity in knots by 22·6. If the heaviness of air be taken at ·077 lb. to the cubic foot on an average, the result is the following approximate rule:—

Divide the square of the velocity of the wind in knots by 150, for the direct impulse on a flat surface in lbs. on the square foot.

Seamen are enabled by experience to refer winds, without measurement, to a series of classes, which have arbitrary names and numbers given to them.

The following Table shows the direct impulse in lbs. on the square foot corresponding to a series of velocities in knots; and also the arbitrary numbers and names given to winds of different

strengths according to the system of Admiral Beaufort (at present used in the British Navy); and according to that of the Swedish Admiral Chapman, described in his treatise on finding the Area of Sails (published in English by the Society for the Improvement of Naval Architecture, 1794).

TABLE OF WINDS.

Velocity. Knots.	Direct Impulse in lbs. on the square foot.	Beaufort.	Numbers and Names. Chapman.
1	·0067	No	1. Light air.
2	·027		
3	·060		
4	·107	2.	Light wind.
5	·167		
6	·240	3.	Light breeze.
7	·327		
8	·427	4.	Moderate breeze.
9	·540		
10	·667	5.	Fresh breeze. Top-gallant-sail wind.
11	·807		
12	·960		
13	1·13		
14	1·31	6.	Strong breeze. Top-sail wind.
15	1·50		
16	1·71		
17	1·93		
18	2·16	7.	Moderate gale. Close-reefed-topsail wind.
19	2·41		
20	2·67		
22	3·23		
24	3·84	8.	Fresh gale. Courses-and-driver wind.
26	4·51		
28	5·23		
30	6·00		
32	6·83	9.	Strong gale. Half storm.
34	7·71		
36	8·64		
38	9·63		
40	10·7	10.	Heavy gale. Whole storm.
45	13·5		
50	16·7		
60	24·0		
70	32·7	12.	Hurricane.
80	42·7		
90	54·0		
100	66·7		

The terms used by Chapman had reference to the power of ships of the line to carry sail, according to his opinion and the practice of his time, and are not to be taken as establishing universal rules on that subject.

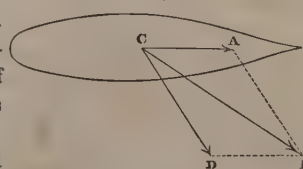
The greatest direct impulse of wind hitherto observed in Britain was 55 lbs. on the square foot; but it probably lasted for an instant only.

According to the experiments of M. Thibault, the impulse of the wind upon a sail, of the usual concave figure, is very nearly equal to its impulse on an *equal area* of a plane surface.

179. *Real and Apparent Motion of Wind.*—By the *real* motion of the wind, is meant its motion relatively to the earth; and by the *apparent* motion, its motion relatively to the ship. It is upon the apparent motion only that the action of the wind on the ship's sails depends.

The motion of the wind relatively to the ship is the resultant of the real motion of the wind, and of a motion equal and directly opposite to that of the ship. Thus, in Fig. 9, let *CB* represent the direction and velocity of the real motion of the wind,

Fig. 9.



and \overline{CA} the direction and velocity of the motion of the ship: through B draw \overline{BD} parallel and equal to \overline{AC} ; then \overline{CD} will represent the direction and velocity of the apparent motion of the wind.

A vane on board ship indicates the apparent direction of the wind, and an anemometer on board ship its apparent force.

It is evident that the strength of the apparent wind may be either greater or less than that of the real wind, according as \overline{CD} is greater or less than \overline{CB} .

[The algebraical expression of these principles is as follows:—

Let $c = \angle CDB$ denote the angle made by the point from which the *apparent* wind blows with the course of the ship;

$k =$ supplement of $\angle CBD$, the corresponding angle for the real wind;

$m = \frac{\overline{CD}}{\overline{DB}}$, the ratio of the velocity of the apparent wind to that of the ship; so that m^2 is the ratio of the direct impulse of the apparent wind to that of a wind blowing with the speed of the ship;

$n = \frac{\overline{CB}}{\overline{DB}}$, the corresponding ratio for the real wind; so that n^2 is the ratio of the direct impulse of the real wind to that of a wind blowing with the speed of the ship:

Then the following cases may occur:

I. Given c, m .

$$n = \sqrt{(1 + m^2 - 2m \cos. c)}$$

(when c is obtuse, + is to be put instead of —);

$$\sin. k = \frac{m}{n} \sin. c.$$

II. Given c, n .

$$m = \sqrt{(n^2 - 1 + \cos.^2 c) + \cos. c}$$

(when c is obtuse, — is to be put instead of + before the last term);

$$\sin. k = \frac{m}{n} \sin. c.$$

III. Given k, n .

$$m = \sqrt{(1 + n^2 + 2n \cos. k)}$$

(when k is obtuse, — is to be put instead of + before the last term);

$$\sin. c = \frac{n}{m} \sin. k.]$$

180. *Oblique Impulse of Wind on Sails.*—Agreeably to the second law of motion and to the principle of action and reaction, the pressure exerted by the wind on a sail is equal in amount and opposite in direction to the change of the momentum of the particles of air which the action of the sail upon them produces in each second.

The direction of that change of momentum is perpendicular, or nearly perpendicular, to the surface of the sail; and its amount is the product of the following two factors:—

The component of the wind's apparent velocity in a direction perpendicular to the sail, divided by gravity; that is to say—

Apparent velocity in feet per second $\times \sin.$ angle of wind and sail;
32.2

The weight of air acted upon by the sail in a second; being the product of the apparent velocity, the heaviness, and the sectional area of the stream of air acted upon.

Hence it appears that the oblique impulse is equal to the direct impulse due to the same apparent velocity of wind multiplied by the sine of the angle made by the apparent motion of the wind

and the plane of the sail, and by the ratio which the area of wind acted upon during the oblique impulse bears to the area of wind acted upon during the direct impulse.

The law according to which the last-mentioned ratio varies with the angle of incidence, has never yet been exactly determined. That ratio has been assumed by some of the early writers on the subject, to be equal to the sine of the angle of the wind and sail, so that the impulse of a given wind on a given sail varies as the square of the sine of the angle between the wind and the sail.

It is certain, however, that the impulse diminishes much more slowly than the square of the sine of the angle, as is proved by the speed with which vessels can sail “close-hauled;” that is, in a direction making the smallest practicable angle with the direction from which the wind blows (being in ordinary cases, from five to six points). The speed of sailing close-hauled, however, is not altogether due to an increase in the pressure beyond that corresponding to the square of the sine; it is to a great extent the effect of the increased strength of the *apparent wind*, when the vessel's course lies near the wind. (Article 179.)

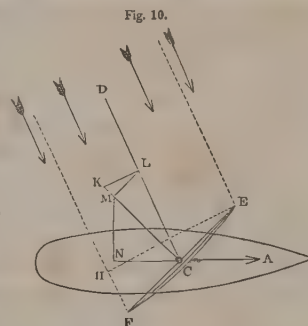
The effort which drives the ship ahead, and that which heels her over and drives her to leeward, are the components, respectively parallel and perpendicular to the keel, of the normal pressure of the wind on the sail.

Let Fig. 10 represent a plan of a ship, advancing in the direction CA ; DC the *apparent* direction from which the wind blows; and EF a sail. From F draw FH parallel to CD ; and from E draw EH perpendicular to CD , cutting FH in H . Then if EH be considered to represent the breadth of the stream of wind which has its momentum in a direction perpendicular to the sail taken away, the area of that stream will vary proportionally to $\sin. DCE$; but the fact appears to be, that when DCE is nearly a right angle, the breadth of that stream is less than EH ; and when DCE becomes very acute, that breadth becomes considerably greater than EH .

Assuming, however, for the sake of illustration, that EH is the breadth of the stream of wind deflected, the action of the wind may be represented as follows:—

Perpendicular to FE draw \overline{CK} , of a length to represent the pressure which would be produced by the direct impulse of the wind on the sail. From K let fall KL perpendicular to CD , and from L let fall LM perpendicular to CK ; then \overline{MC} will represent the normal pressure or total effort on the sail. From M let fall MN perpendicular to AC ; then \overline{NC} will represent the forward or longitudinal component of that effort, which produces headway; and \overline{MN} the transverse component, which produces lee-way, and makes the ship lean over to leeward.

One of the consequences of the assumption that the normal pressure varies as the square of the sine of the angle of incidence, is the following: that the greatest forward effort on a given sail for a given apparent direction of the wind takes place when the sail is braced so that the tangent of the angle which it makes with the apparent wind, is double of the tangent of the angle which it makes



with the ship's course. The position of the sail which fulfils that condition is found by the following construction:—In Fig. 11, let

PCA represent the direction of the ship's course. On that line describe two semicircles touching each other at A, such that the diameter, PA, of the larger semicircle shall be three times the diameter, CA, of the smaller. Draw CD towards the point from which the wind apparently blows, cutting the larger semicircle in D; join DA, cutting the smaller semicircle in Q; CQ will be the required position of the sail.

Considering that the pressure of the wind on a sail does not vary precisely as the square of the sine of the angle of incidence; that sails are not flat, but more or less concave; and that the direction, extent, and velocity of the stream of wind which acts on a sail are often more or less changed by the action of other sails to windward of it—it is only natural that the results of practice should in many cases differ from those given by the preceding rule.

The agreement of the rule with practice is best in the case of fore-and-aft rigged vessels; probably because their sails are on the whole flatter than those of square-rigged vessels, and interfere less with each others' action on the wind.

A sail which bellies out so as to present a considerable concavity to the windward side, is most efficient when the yard is braced more square than the angle assigned by the rule; that is, when making a greater angle with the ship's course, and a less angle with the wind.* When the sails of a vessel sailing near the wind assume a very concave form, it is usually found in practice that their most effective position is with the "weather leech," or edge next the wind, just on the point of shivering—that is, edgeways towards the wind; while the "lee leech," or edge from which the wind finally glances off, conforms in its position approximately to the rule already stated.

The results of experience in the propulsion of vessels by sails agree in a general way with the law assumed by Chapman in his treatise on that subject—viz., that *when the sails are braced to the most efficient angle (an angle to be found by practical trials), the forward effort of the wind varies proportionally to the square of the velocity of the apparent wind, and to half the versed sine of the angle between the ship's course and the direction from which the apparent wind blows.*

That theory is expressed geometrically as follows:—

In Fig. 11A, let CA represent the vessel's course and speed; DA the direction and velocity of the real wind; then DC will represent the direction and velocity of the apparent wind. In CA (produced if necessary) take CE = CD, and join DE. Then—

As $4 \overline{CA}^2$

: is to \overline{DE}^2

: : so is the effort due to the direct impulse, on a given area of sail, of a wind blowing with a speed equal to that of the vessel,

: to the actual forward effort with the same area of sail.

The inverse ratio, $\frac{4 \overline{CA}^2}{\overline{DE}^2}$, is proportional to the area of sail required to drive the vessel with a given speed.

The effort thus determined is equal to that which would be exerted if a stream of wind, of a sectional area equal to *one-half* of the area of canvas spread, were to be deflected so as to glance off from the sails in a direction due aft.

[The following Table gives a few examples of the results of that supposition.

The speed of the ship is taken as the unit of velocity, and the direct impulse of a wind blowing with that velocity as the unit of force;

$n = \frac{\overline{DA}}{\overline{CA}}$ denotes the ratio of the velocity of the *real wind* to

that of the ship; it is assumed successively to be 3, 2, and $1\frac{1}{2}$;

$k = \angle DAE$ denotes the angle made by the direction from which the *real wind* blows with the course of the ship;

$m = \frac{\overline{DC}}{\overline{CA}}$, and $c = \angle DCE$ denote the corresponding ratio and angle for the *apparent wind*, calculated by the formulæ of Article

179;

$\frac{m^2 \text{ versin. } c}{2} = \frac{\overline{DE}^2}{4 \overline{CA}^2}$ denotes the *proportionate forward effort per square foot of canvas*, according to Chapman's theory;

$\frac{2}{m^2 \text{ versin. } c} = \frac{4 \overline{CA}^2}{\overline{DE}^2}$ denotes the *proportionate area of canvas* required to drive the vessel at one-third, one-half, or two-thirds of the speed of the real wind, as the case may be.]

REAL WIND.....				n = 3.				n = 2.				n = 1½.							
				APPARENT WIND.															
k				m		c		m² vers. c. 2		2 m² vers. c.		m		c		m² vers. c. 2		2 m² vers. c.	
Points.																			
4	=	45°	0'	3.77	34° 12'	1.28	.81
5	=	56°	15'	3.65	43° 4'	1.80	.56	2.69	38° 14'	0.77	1.29
6	=	67°	30'	3.51	52° 15'	2.38	.42	2.56	46° 19'	1.01	0.99	2.10	41° 25'	.55	1.82
8	=	90°	0'	3.16	71° 34'	3.42	.29	2.24	63° 26'	1.38	0.72	1.80	56° 18'	.72	1.38
10	=	112°	30'	2.78	93° 2'	4.05	.25	1.86	82° 45'	1.52	0.66	1.45	72° 56'	.74	1.35
12	=	135°	0'	2.40	117° 53'	4.23	.24	1.47	106° 21'	1.40	0.71	1.06	93° 26'	.60	1.67
16	=	180°	0'	2.00	180° 0'	4.00	.25	1.00	180° 0'	1.00	1.00	0.50	180° 0'	.25	4.00
(1)				(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)				

The results of calculation in the preceding Table agree in a general way with those of practical experience.

* The remarks on this point of Robison (who was himself a seaman) are worthy of attention. —(Encyc. Brit., Article SEAMANSHIP.)

Thus, a comparison between columns (4), (8), and (12) illustrates the fact, that while a small area of canvas produces its greatest forward effort with the wind on the quarter, or nearly astern, a large area of canvas produces its greatest forward effort with

the wind nearly abeam. The reason for that fact appears in columns (2), (6), and (10), which show that the proportionate variation in the strength of the apparent wind is greater, when the speed of the ship is a large fraction of that of the real wind, than when it is a small fraction.

181. *Centre of Effort—Centre of Lateral Resistance—Ardency and Slackness.*—The term "Centre of Effort" is applied to the point in the vertical longitudinal middle plane of the vessel, which is traversed by the resultant of the pressure of the wind on the sails. Its position varies according to the quantity of sail spread, and the manner in which the sails are braced or trimmed; but for purposes connected with the designing of the sails of a ship, a definite approximate position is found for it as follows:—"All plain sail"—that is to say, all the sails more commonly used, and which can be carried with safety in a "Fresh breeze," No. 5, are supposed to be set (as to those sails, details will be given in the Fifth Division); they are drawn on a plan, being a vertical side elevation of the vessel, as if they were braced exactly fore and aft, or in a vertical longitudinal plane; and their common centre is found by the rules of Chapter II., Article 37.

By the "Centre of Lateral Resistance" is meant, the point in the longitudinal midship plane of the vessel, which is traversed by the resultant of the resistance of the water to the motion of the vessel. No method is yet known by which it can be determined from theory. Its position longitudinally, relatively to the centre of effort in an actual vessel, may be inferred from the ardency or slackness shown by that vessel: for ardency indicates that the centre of effort is abaft the centre of resistance; and slackness, that the centre of effort is afore the centre of resistance.

It is thus known that in ship-rigged vessels and barques the centre of lateral resistance is from one-twentieth part to one-sixtieth part of the length of the ship afore the centre of the vertical longitudinal section (the former limit having reference to long ships, the latter to short ships); for such is the position of the centre of effort, in those vessels, that gives a proper medium between ardency and slackness.

In schooners and cutters, the centre of effort of the sails in good examples has been found to be sometimes a little afore and sometimes a little abaft the centre of the vertical longitudinal section; so that this point itself probably coincides nearly with the centre of lateral resistance.

A large amount of deadwood abaft causes the centre of lateral resistance to be well aft, requiring the centre of effort to be well aft also. A long sharp gripe or forefoot, on the other hand, carries the centre of lateral resistance well forward, and tends to produce ardency, unless the centre of effort of the sails be carried sufficiently far forward also. Precise rules on this subject cannot yet be laid down for want of sufficient data, but the above will be sufficient for general guidance. The naval architect will, in designing a ship, fix the centre of effort of the sails in accordance with the above rules; when the ship has been tried at sea, he will learn her qualities, and will thus obtain for himself a rule for his guidance in designing similar ships.

182. *Area and Moment of Sail—Stiffness under Sail.*—The "Moment of Sail" is a measure of the tendency of the wind to heel the vessel over, and is computed according to the supposition stated in the last Article, viz.: that "all plain sail" is set, and braced exactly fore and aft. The pressure is also assumed to be one pound on the square foot. Thus the moment

of sail is the product of two factors—the area of sail, and the height of the centre of effort above the centre of lateral resistance; that being the arm of the heeling couple formed by the lateral pressure of the wind and the lateral resistance of the water.

In almost all published calculations of the moment of sail, the arm of the couple is measured from the plane of flotation, and therefore requires to be increased by the depth of the centre of lateral resistance below that plane; a depth whose precise value is uncertain, but which may be estimated at about one half of the draught of water amidships. The effect of this is to add one-eighth or thereabouts to all moments of sail which are computed relatively to the plane of flotation only.

In vessels provided with all the sail that they can conveniently carry, the breadth of the sails cannot go beyond a certain proportion to the length of the ship, nor can their height exceed a certain proportion to her breadth. Hence the area of sail, in similar vessels, is roughly proportional to the product of the length and extreme breadth; the height of the centre of effort above the centre of resistance is roughly proportional to the extreme breadth; and the moment of sail is roughly proportional to the product of the length and square of the breadth; which product is nearly proportional to the tonnage by old measurement.

From examples given by Mr. Fincham in his treatise on the Masting of Ships, it appears that in different classes of vessels the following proportions are borne by the leverage of sail (or height of the centre of effort above the centre of resistance) to the extreme breadth, and by the area of sail to that of the plane of flotation.

	Leverage of Sail ÷ Extreme Breadth.	Area of Sail ÷ Area of Plane of Flotation.
Boats,.....	1.0 to 1.8	2.0 to 4.0
Cutters,.....	1.5 to 1.7	3.2 to 3.6
Schooners,.....	about 1.7	3.6 to 5.0
Brigs,.....	1.7 to 2.0	3.5 to 3.75
Corvettes and Frigates,.....	1.75 to 1.9	3.1 to 3.9
Ships of the Line,.....	1.8 to 1.9	2.9 to 3.25
Ordinary limits for Decked Vessels,	1.5 to 2.0	3.5 to 4.5

The only accurate method, however, of adapting a vessel's sails to her power of carrying sail, is to make the moment of sail equal to the moment of stability at a definite angle of heel. Such was the method followed by Chapman, in his treatise on the Sails of Ships of the Line, before referred to. He assigned 7° as the greatest angle of steady heel admissible in those ships while fighting their lee guns, of which about 2½° were due to the weight of the men at the lee broadside, and the remaining 4½° to a moment produced by a lateral pressure of about one pound per square foot of canvas. The following statement of the angles of heel corresponding to the moment of sail in different classes of vessels is computed partly from data contained in Mr. Fincham's work already referred to, and partly from other recent examples:—

	Angle of heel per lb. of lateral pressure per square foot of canvas.	Circular Measure.
Ships of the Line, Frigates, and large Merchant Ships, }	4°	0.070
Corvettes,.....	5°	0.087
Schooners and Cutters,.....	6°	0.105
Yachts {from	6°	0.105
{to.....	9°	0.157

From such data as these, the greatest moment of sail suitable to a given vessel can always be computed as follows:

$$\begin{aligned} \text{Moment of Sail} &= \text{Displacement in tons} \times 2240 \\ &\times \text{Height of Metacentre above Centre of gravity} \\ &\times \text{Angle of heel in circular measure;} \end{aligned}$$

and by the same rule it may be determined, whether a vessel has sufficient stability to carry a given proposed quantity of sail.

183. *Speed under Sail.*—From what has been stated in Articles 157 and 166 it appears, that if the figure and dimensions of a ship are properly adapted to her intended speed, her resistance is equal to the square of the speed in knots, multiplied by about $\frac{1}{100}$ th of a lb. per square foot of *augmented surface*, for a clean iron bottom, and probably about a tenth part less for copper sheathing.

From Article 178 it appears, that the direct impulse of the wind is equal to the square of its velocity in knots, multiplied by about $\frac{1}{100}$ th of a lb. per square foot of surface.

Hence if the *forward effort* on a vessel's sails is of an intensity equal to that of the direct impulse of a wind having the same speed with the vessel, the area of canvas to produce that speed should be from 1.35 to 1.5 times the *augmented surface of the ship*; and if the intensity of the forward effort has a different value, bearing a given ratio to the above, the area of canvas is to be altered in the inverse ratio.

For example, the numbers in columns (5), (9), and (13) of the Table near the end of Article 180, being multiplied by 1.5, give approximately the ratios of area of canvas to augmented surface suitable for driving an iron ship at one-third, one-half, and two-thirds of the velocity of the real wind respectively, when her course makes with the point from which the real wind blows, the angles set down in column (1).

It can also be deduced approximately from the Table, what proportion of the speed of the ship to that of the real wind may probably be expected from certain given proportions of canvas to augmented surface, on given points of sailing; for example:—

Ratio of Area of Canvas to Augmented Surface.	Relation between Course and Wind.	Probable ratio of Speed of Ship to Speed of Real Wind.
1.....	{ Course 5 points near wind,.....	$\frac{1}{3}$
	{ Wind 2 points abaft beam,.....	$\frac{1}{3}$
1½.....	{ Course 6 points near wind,.....	$\frac{1}{3}$
	{ Wind abeam,.....	$\frac{2}{3}$
	{ Wind astern,.....	$\frac{1}{2}$
2.....	{ Course 5 points near wind,.....	$\frac{1}{3}$
	{ Wind 2 points abaft beam,.....	$\frac{2}{3}$
2½.....	{ Course about 6½ points near wind,.....	$\frac{1}{3}$
	{ Wind on quarter,.....	$\frac{2}{3}$

The ordinary ratios of area of plain sail to augmented surface range from equality to nearly double. A common proportion in large merchant ships is about 1.5.

184. *Speed under Canvas and Steam Combined.*—One effect of the combination of sail with steam-power in propelling a ship, is to increase the efficiency of the propeller; for as it has a part instead of the whole of the resistance of the water to overcome, its slip is diminished.

When such increase of efficiency is neglected, the probable effect of combining steam-power and canvas is deduced approximately as follows; and the error is on the safe side:—

The work done in a given time by the steam-power is proportional to the cube of the speed under steam alone:

The work done in the same time by means of the sails is nearly proportional to the cube of the speed under canvas alone.

The work done by the sail and steam-power combined is proportional to the cube of the actual speed of the vessel; therefore that cube is nearly equal to the sum of the cubes of the speed under steam alone and under canvas alone.

TABLE OF EXAMPLES.

Speed under canvas ÷ speed under steam.	Probable speed under steam and canvas ÷ speed under steam.	Speed under canvas ÷ speed under steam.	Probable speed under steam and canvas ÷ speed under steam.
4.....	1.02.....	1.3.....	1.47.....
5.....	1.04.....	1.4.....	1.55.....
6.....	1.07.....	1.5.....	1.64.....
7.....	1.10.....	1.6.....	1.72.....
8.....	1.15.....	1.7.....	1.81.....
9.....	1.20.....	1.8.....	1.90.....
1.0.....	1.26.....	1.9.....	1.99.....
1.1.....	1.33.....	2.0.....	2.08.....
1.2.....	1.40.....		

In vessels whose chief propelling power is that of steam, the area of canvas spread is in most cases proportionally small, ranging from one half of to once the augmented surface.

185. *Figures and Proportions of Sailing Vessels.*—In a sailing vessel, as well as in a steamer, smallness of resistance depends on the absence of “supernumerary diverging waves” and on smallness of augmented surface. But the propelling power which a sailing vessel can carry, unlike that of a steamer, depends not upon her displacement, but upon her stability; and consequently, while speed in a steamer is promoted by giving her the least possible augmented surface compared with her displacement, speed in a sailing vessel is promoted by giving her the least possible augmented surface compared with her stability.

Hence it is that the best examples of sailing vessels are broader for their length than the best examples of steamers. The common proportions of length to breadth in the navies of the world down to recent times, ranged from about 3 to $3\frac{3}{4}$; and 4 was considered a great proportion, even in yachts. At present many sailing vessels are built longer and narrower than those proportions; but still, in the finest examples of clipper ships, the length is seldom more than from 5 to 6 times the breadth.

While a certain degree of stiffness is necessary to enable a vessel to carry sail, excessive stiffness, and in particular an increase of stiffness as the ship heels over, may endanger the carrying away of masts by a sudden gust of wind. Hence it is advantageous, in a sailing vessel, to use between wind and water the isochronous form of cross section described in Article 126A, which gives equal stiffness at all angles of heel; but this is seldom attended to.

It is necessary, in order to prevent excessive leeway, that a sailing vessel should meet with great resistance to moving sidewise through the water. That object is promoted by a sufficient draught of water, a sharp floor, and a deep keel. In long vessels with fine ends, the sharpness of form of the cross sections near the ends is held to compensate to a certain extent for roundness or flatness amidships.

The combination of a comparatively broad plane of flotation with fine lines below water, is effected by making the angle of entrance, or obliquity of the water-lines, diminish progressively from the load-water-line downwards; so that in vessels (such as yachts) built for quick sailing alone, irrespective of other qualities, the permanently immersed parts of the cross sections are nearly triangular. When capacity and carrying power, as well as speed, are aimed at, the cross sections amidships are made of a figure suitable for those objects, as rectangular or elliptical, while those near the ends are triangular or nearly so.

Should the “flaring” or outward spreading of the cross sections near the bow and stern be so great as to tend to produce vertical motion of the ship's centre of gravity to any material extent (as explained in Article 112), that tendency may be corrected by

means of a comparatively small inward inclination, or "tumbling home" of the midship section (see Article 113).

In every vessel whose cross sections between wind and water are not arcs of circles with their centres in her longitudinal axis, the heeling of the vessel alters the form of the water-line. In consequence of the flaring out of the vertical cross sections near the bow and stern, that alteration of form usually consists in becoming fuller at the lee side and finer at the weather side; and this, by increasing the pressure of the water against the lee bow, is favourable to weatherliness and arduency. But it also tends to increase the resistance; and therefore it should not be greater than is necessary in order to give sufficient sharpness of form to the vertical sections near the ends of the vessel.

Small flat-bottomed vessels for sailing in shallow water, when the draught of water is only $\frac{1}{4}$ th or $\frac{1}{5}$ th of the breadth of the vessel, or even less, are often provided with moveable boards, which are lowered edgewise into the water, so as to resist leeway in the manner of a keel. These are called "lee boards" or "centre boards," according as they are lowered into the water at the lee side of the vessel, or through a well amidships.

Further remarks on the forms of sailing vessels, as affecting their "handiness," or manœuvring qualities, will be given in the next Section.

SECTION VI.—HANDINESS.

186. *General Explanations.*—By "*handiness*" in a ship is meant the property of being easily, quickly, and accurately manœuvred, the manœuvres consisting mainly in changes of the direction in which the ship moves, or in which her head points. Such changes are produced by the action of the rudder on the water, by that of propelling apparatus, by that of sails, or by those actions combined in various ways. The performance of the various manœuvres of which a vessel is capable belongs wholly to seamanship; the present treatise is limited to the consideration of the forms and proportions which are favourable to that performance.

187. *Action of the Rudder.*—By the *Rudder* is meant that part of the steering apparatus which is wholly or partly immersed in, and directly acted on by, the water. The *whole* steering apparatus, including the rudder, together with the mechanism by which it is moved, is comprehended under the word *Helm*.

The immersed part of the rudder has usually the form of a flat plate in a vertical or nearly vertical position, capable of turning into different angular positions about a vertical or nearly vertical axis. In the old and still most common form of rudder, that axis is at the forward edge of the rudder, where it is hinged by means of what are called "pintles" and "braces." In a form called the *balance-rudder*, now sometimes used, the axis of motion is at about *one-third* of the breadth of the rudder from its forward edge; and the rudder turns on a pivot at its lower end, where it is supported by the projecting after end of the keel. In a third form of rudder (introduced by Mr. Lumley), the rudder has a hinge in the middle of its breadth, dividing it into two parts called the "body" and the "tail;" and by means of suitable mechanism, the tail is put over to a greater angle than the body.

The remarks on the rudder in the present Section will be confined wholly to its immersed part, leaving the details of the construction of steering apparatus to be described in the Fourth Division. Some reference, however, must be made to those details,

in order to explain the terms used by seamen in describing the positions of the rudder. Those terms are founded upon the positions taken by the *tiller*, which is a horizontal lever usually projecting directly forwards from the rudder-head, and therefore pointing exactly in the opposite direction to that in which the after edge of the rudder points; and the same terms are used whether the ship actually has a tiller, or some other means of moving the rudder. Hence the statement that the helm is inclined or "put over" in a given direction, means that the rudder is inclined to the ship's keel in the opposite direction. For example—

HELM	Means, that the RUDDER is
<i>A-starboard</i> , or inclined towards the right;	<i>A-port</i> , or inclined towards the left;
<i>A-port</i> , or inclined towards the left;	<i>A-starboard</i> , or inclined towards the right;
<i>A-lee</i> , or inclined to leeward;	<i>A-weather</i> , or inclined to windward;
<i>A-weather</i> , or inclined to windward;	<i>A-lee</i> , or inclined to leeward.

In order that the rudder, when acting alone, may have power to turn or guide the ship, it is necessary that the ship should have "*steerage-way*;" that is, should be moving through the water with such a speed, as to make the particles of water act on the rudder with a pressure sufficient to turn the vessel.

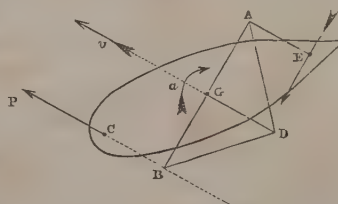
It is also necessary that the rudder should be immersed, not in a mass of eddies dragging behind the ship, but amongst particles of water whose motion, *relatively to the ship*, consists in a steady flow astern. Hence *the same fairness and fineness of the water-lines and buttock-lines of the after-body which are essential to speed and economy of power* (as explained in Articles 159 and 160), *are essential to good steering also.*

When the helm is put over, the rudder deflects the motion of the stream of particles of water which it meets, so as to cause those particles to glance off the rudder in the direction of a tangent to its after edge. Through the reaction of those particles, a pressure is exerted on the rudder in a direction perpendicular, or nearly perpendicular, to its surface. That pressure may be resolved into a longitudinal and a transverse component; the longitudinal component simply increasing the resistance to the motion of the vessel, and so diminishing her speed; and the transverse component turning her round, and at the same time driving her sidewise, in the direction opposite to that in which the rudder is inclined.

The *first* action of the rudder, on being put over, takes place agreeably to the principles explained in Article 79. Let Fig. 12 represent a deck-plan of

Fig. 12.

a vessel whose helm has been put over to port, so that her rudder is inclined to starboard. Let CP represent the pressure exerted by the water normally to the rudder; that pressure causes the existing progressive motion of the ship to be combined with a rotation about an instantaneous axis A, determined as in Article 79; and that rotation may be resolved into a rotation in the direction shown by the arrow (that is, to starboard) about the ship's centre of gravity, G, combined with a translation in the direction, *v*; which translation consists partly in retardation of the vessel's speed, and partly in drifting sidewise to port. That sideward drifting increases until the lateral resistance of the water becomes equal to the transverse component of the pressure on the rudder; and then the vessel goes on turning under the action of a



couple of forces, whose *arm* or lever is the distance between the centre of the rudder and the *centre of lateral resistance*, already mentioned in Article 181; and the longer that lever, the more efficient is the action of the rudder.

Now it is well established that the centre of lateral resistance in almost every case lies *before* the middle of the vessel's length; therefore the *ordinary position of the rudder, at the stern, is the most efficient of all possible positions*; and hence the failure of all attempts to improve its action by placing it elsewhere.

From the same principle it follows, that a raking stern-post, by bringing the centre of the rudder forward, diminishes its efficiency.

The angular velocity with which the vessel turns is that at which the moment of the resistance of the water to her turning is equal to the moment of the couple just mentioned. That resistance is greatest in long vessels, and in those that are sharp fore and aft; and hence such vessels turn slowly.

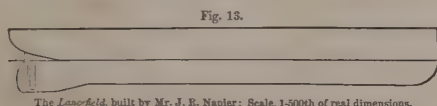
The efficiency of the rudder is increased in steamers, by putting it as far over as is practicable with the force at command, driving the vessel at full power round a circle, and observing the diameter of the circle, and the time occupied in describing it.

In ordinary cases, the diameter of the circle is about six or six and a half times the length of the vessel.

From the results of a number of experiments collected and arranged by Mr. N. Barnaby (in the Transactions of the Institution of Naval Architects for 1863) it appears that the two ends of a steamer under full power turn *relatively to the centre of her length*, with a speed which ranges in various vessels from 0.9 knot to 1.5 knot, the mean of all the experiments giving 1.24 knot. Hence it is easily computed, that the time occupied by those vessels in turning completely about, ranged from $1\frac{1}{2}$ second to 2 seconds per foot of length, and was on an average $1\frac{1}{2}$ second per foot of length.

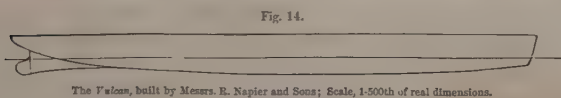
The efficiency of the rudder is increased by giving it a great depth as compared with the mean draught of the vessel, and such is the object of giving fore-and-aft rigged vessels a much deeper draught of water aft than forward. The same figure is not suitable for square-rigged ships, because of the difficulty in them of bringing the centre of effort sufficiently far aft.

The same object (combined with that of giving a deeper immersion to the screw) is attained in the iron screw steamer *Lancefield*, represented in Fig. 13, by means of a curved



depression at the after end of the keel; so that while the vessel floats on an even keel, with a uniform draught of 8 feet, the depth of immersion of the rudder and screw-aperture is $10\frac{1}{2}$ feet. This vessel has been found to be remarkably easy to steer through intricate channels, and to sail well under canvas and steam combined.

The resistance of the water to the turning of a vessel is dimin-



ished by cutting away the forefoot and dead-wood as much as possible; but this cannot be carried far without injury to her

weatherliness, and to her steadiness in a sea-way. Where those qualities are not required, it has been practised with complete success; for example, in the river-steamer *Vulcan*, Fig. 14.

188. *Dimensions and Figure of Rudder*.—The depth of the rudder is regulated by the depth of immersion of the stern-post.

If it be assumed that the normal pressure of the water upon the rudder is proportional to the square of the sine of the angle of incidence of the water upon it, it can be shown that when a *given statical moment* is exerted in putting the helm over, the force which turns the vessel will be greatest if the breadth of the rudder is such that the given moment is just sufficient to force it over to such an angle as makes the angle of incidence of the water 45° . The particles of water which meet the rudder are always in the act of converging slightly towards the vessel; hence the most effective angle of obliquity for the rudder must be rather less than 45° ; and in fact, experience shows it to be 42° or thereabouts. In very large vessels, however, there is seldom force enough available to put the helm over so far.

The *extreme breadth* of the rudder, according to the old practice of shipbuilding, was almost always *one-thirtieth* of the length of the ship on the load-water-line.^o In steamers, according to present practice, it varies from *one-fortieth* to *one-fiftieth* of the length, nearly; and in one example (the *Great Eastern*) it is about *one-eighthieth*.

When the helm is put over, the pressure of the water upon it tends to make the vessel heel, unless the centre of the rudder is on a level with the centre of lateral resistance. In vessels which are to carry much sail this principle should be attended to: in those where chief propelling power is steam, it is of less importance.

It is the practice to round the upper and lower after corners of the rudder, so that its *mean breadth* is from $\frac{3}{4}$ ths to $\frac{4}{5}$ ths of its extreme breadth. Several examples of this are given in the Plates which illustrate this treatise.

In sea-going ships, it is important to the safety of the rudder, that its broad part should be constantly under water, and should not be uncovered when the vessel pitches.

From experiments by Mr. J. R. Napier on the s.s. *Messina* it appeared, that the statical moment required to put the helm over to 40° was equal to the product of the following factors; the area of the rudder in square feet; the distance of the centre of that area from the axis of motion, in feet; the square of the speed of advance of the screw, in knots; and the constant '94. When the square of the speed of the ship was used instead of that of the screw, the constant multiplier was 1.4.

The *Balanced Rudder* requires a comparatively small force to put it over; but experience of its use has not yet been extensive enough to enable rules for its dimensions to be laid down, except as to the position of its axis. In order that a balanced rudder may tend of itself to come to a fore-and-aft position, it is necessary that the lesser part of its breadth should be before, and the greater abaft, its axis of motion. The proportions of $\frac{1}{3}$ before and $\frac{2}{3}$ abaft, have been used with success. When the axis is at the middle of the breadth, the rudder tends to place itself athwartships, and to remain there.

As a substitute for a rudder in flat-bottomed vessels for shallow-

^o That eminent shipbuilder, the late Mr. John Wood, made the breadth of the rudder $\frac{1}{4}$ th of that of the ship.

water navigation, there has sometimes been used a pair of vertical sliding boards, making angles of 45° to starboard and to port of the keel respectively, and capable of being lowered into and drawn out of the water through a pair of narrow vertical wells in the quarters of the vessel; the amount of steerage being regulated by the extent of the area of board immersed.

189. *Manœuvring by Engine-Power.*—The action of the rudder in every steam vessel is more or less modified through the motion impressed on the particles of water by the propelling apparatus—the effect on the whole being, to make the rudder somewhat more powerful than in a sailing vessel. That effect is greatest in a screw steamer, because of the closeness of the screw to the rudder. Even when the rudder is amidships, the rotation of the screw has a slight tendency to turn the vessel—to starboard, if the screw is right-handed; to port, if it is left-handed: and the vessel takes less time to describe a circle to starboard or to port, according as her screw is right or left handed.

With a pair of *Twin Screws* (that is, screws driven by two independent engines) a vessel can be turned about on her own centre, by working one screw ahead and the other astern. From experimental trials on vessels in which the transverse distance between the axes of the screws is about $\cdot 06$ of the length of the vessel, it appears that when the helm is put hard over at the same time that the screws are thus worked, the time occupied in turning about differs but little from that of describing a circle.

It is obvious that where the distance between the centres of the screws can be increased proportionally to the length of the vessel, the time of turning on her centre will be shortened.

Paddle Wheels driven by independent engines have long been used in American river-steamers, and in tug-steamers in Britain.

It has often been proposed to employ a *manœuvring screw*—that is, a small screw-propeller with its axis athwartships, placed in a round aperture either in the dead-wood or the gripe.

In a twin passenger-steamer,^a *manœuvring paddle wheels* at the bow and stern, with their axes lying fore and aft, have been used.

Another method of manœuvring by steam-power consists in connecting the screw with its shaft by an universal joint, so that the axis of the screw can be turned aside to various angles along

with the rudder. In order that the screw, when in an oblique position, may rotate with an uniform angular velocity, a double universal joint should be used.

Vessels propelled on Mr. Ruthven's plan, by ejecting jets of water, are manœuvred by changing the directions of the nozzles of the jets.

190. *Manœuvring by Sail* is effected by trimming different sails, or sets of sails, so that the wind shall act upon them with different forces and in different directions. For purposes of manœuvring, the sails are distinguished into *head sail* and *after sail*—head sail comprehending all sails whose centres lie before the general centre of effort of all the sails; and after sail, all sails whose centres lie abaft that point. By “shivering” either of those two sets of sails (that is, placing them edgewise towards the wind), the other set is left to act alone; by “backing” one set, and “filling” the other, the wind can be made to act upon them in contrary directions; and by these, and other changes, a good seaman can make a ship perform a great variety of manœuvres with very little assistance from the rudder.

The capacity which a ship possesses for being thus manœuvred depends on the proportionate areas and moments of the head and after sail, and on the positions of the two separate centres of effort of those two sets of sails.

Fincham, in his treatise on the Masting of Ships, compares together the respective moments of the head and after sail relatively to a vertical axis standing in the middle of the length of the load-water-line; and shows, that in good examples of *square-rigged ships*, the *moment of after sail varies from $\frac{1}{10}$ ths to $\frac{1}{10}$ ths of the moment of head sail*, relatively to that axis; and that in *fore-and-aft rigged vessels*, the moment of after sail varies from once to 1·3 times the moment of head sail.

But what the power of manœuvring by sail must principally depend upon, is the *horizontal distance between the separate centres of effort of the head sail and after sail*; for that distance is the lever at the two ends of which those two sets of sails act in turning the vessel. Its ordinary value, as computed from some practical examples, appears to be from $\frac{1}{10}$ ths to $\frac{1}{10}$ ths of the length on the load-water-line.

CHAPTER VI.

ON THE DESIGNING OF SHIPS.

191. *General Design—Outside Dimensions.*—It would not be desirable, even if it were possible, to lay down an invariable system of rules as to the method and order to be followed in designing a ship. All that can be done in the present Chapter is to indicate, in a general way, the nature of the processes which may be required. The details of those processes, the best way of arranging them, and the omission of some of them, are matters to be decided by the judgment of the naval architect in each particular case.

One of the principal conditions to be fulfilled by a proposed ship almost always is, that she shall be capable of carrying a certain

burden, or load, in tons of 2240 lbs., and shall be of a certain internal capacity in tons of 100 cubic feet. Hence, by the aid of principles explained in Article 99, the naval architect can estimate what her *displacement* ought to be.†

† In addition to the data given in Article 99 as to the ordinary proportions of burden and capacity to displacement, the following results may be given, computed from information as to merchant sailing ships contained in a paper by Mr. John Vernon (Proceedings of the Institution of Mechanical Engineers, August, 1863).

	Per 1000 Tons Register.	
	Iron Ship. Tons.	Wooden Ship. Tons.
Weight of Ship,.....	680	780
Rig, Outfit, Water, and Stores,.....	120	120
Light Displacement,.....	750	900
Cargo,.....	1483	1333
Load Displacement,.....	2233	2233

Z

^a The *Alliance*, designed by Mr. George Mills.

The *Outside Dimensions* of a ship's displacement are, the *length* on the plane of flotation, the *extreme breadth*, and the *mean load draught* of water or *immersed depth* amidships, down to the lower edge of the rabbet, where the skin of the vessel joins the keel. The product of those three dimensions gives the volume of a certain rectangular solid; and that volume, multiplied by a certain *coefficient of fineness* (as explained in Article 97), is equal to the displacement. As a step, therefore, towards determining the three outside dimensions, it is necessary that the naval architect should decide what coefficient of fineness the ship is to have. That will depend on the figures which he intends to adopt for the midship section, and for the water-lines; for (as stated in Article 97) the coefficient of fineness of the displacement is the product of the coefficient of fineness of the midship section, and of the mean coefficient of fineness of the water-sections.

From what has been stated in Article 91A, it appears that a close approximation to the mean coefficient of fineness of all the water-sections is obtained in ordinary cases by taking the coefficient of fineness of a water-section situated at *one-third of the immersed depth* below the load-water-section.

The water-lines of the fore-body and after-body may, and very often do, differ in fineness; and they may or may not have the straight water-lines of a middle body between them. To find the coefficient of fineness of an entire water-section from those of its parts, the naval architect must decide what proportions of the whole length are to belong to the fore-body, the middle-body, and the after-body respectively; then, multiplying each division of the length by its proper coefficient of fineness, and dividing the sum of the products by the whole length, the quotient is the coefficient of fineness of the whole water-section. The coefficient of fineness of the straight middle division is always *unity*.

The displacement in cubic feet being divided by the coefficient of fineness of the whole immersed body, gives *the product of the three outside dimensions*.

In order to find those three dimensions separately, when no conditions are laid down to limit their absolute values, the naval architect must decide what proportions they are to bear to each other. Then multiplying together the proportions which the length and the immersed depth are respectively to bear to the breadth, a divisor is obtained, by which the volume of the rectangular solid is to be divided; and the *cube root of the quotient will be the extreme breadth*.

The ordinary proportions of length to breadth may be taken as ranging from 3 to 7 in sailing vessels, and from 5 to 12 in steamers; those of immersed depth to breadth, from $\frac{1}{2}$ downwards. (See Articles 170, 171, 185.)

In certain cases, limits may be put to the absolute values of the outside dimensions. For example—

I. The *Least Length of fore-body and after-body* consistent with economy of power, at the greatest intended speed of the vessel, may be determined by the principles explained in Article 158.

II. The *Depth of Immersion* may be limited by the shallowness of the water which the vessel is to navigate.

III. The *Extreme Breadth* may be limited by the width of dock gates, or other passages which the vessel is to traverse.

When the quality aimed at in the vessel to be designed is speed under sail, irrespective of other qualities (as in the case of sailing yachts), the primary condition to be fulfilled may be, that the plane of flotation shall have a given area (for to that area the area of sail

is nearly proportional). In this case, the naval architect, so soon as he has fixed the coefficient of fineness of the load-water-section, can at once calculate the area of its circumscribed rectangle, which is the product of the length and breadth. He has next to decide what proportion the length is to bear to the breadth; and then the absolute length and breadth are to be computed. The midship draught of water will then be fixed, according to the form and proportions chosen for the midship section, with a view to weatherliness, handiness, and steadiness. The ordinary proportion of midship depth of immersion to breadth in sailing yachts ranges from $\frac{1}{2}$ to $\frac{3}{4}$. When it is $\frac{1}{2}$, or smaller, lee boards or a centre board are in general required (Article 181).

192. *Keel, Stem, Stern-post, and Rudder*.—The determination of the length and midship draught of water gives the dimensions of the immersed part of the midship longitudinal section of the vessel, bounded by the forward edge of the rabbet of the stem, the lower edge of the rabbet of the keel, and the after-edge of the rabbet of the stern-post; and the naval architect is then enabled to decide upon the figure of that section, with a view to the position of the centre of lateral resistance (Art. 181), and the action of the rudder (Art. 187). He will fix, for example, whether the ship is to float on an even keel, by the stern, or by the head; whether the keel is to be straight or curved, of uniform or of varying depth; whether the stern-post is to be upright or raking; whether the stem is to be straight or curved, upright or raking, &c. The figure and dimensions of the rudder can also be fixed (Article 188).

193. *Moulded Dimensions and Displacement*.—It is more convenient for practical purposes, though less scientifically exact, to design the figure, not of the outer surface of the ship, but of the *inner surface of her skin*; because upon the figure of that inner surface the shapes of all the pieces of the frame directly depend.

The principal dimensions of the inner surface of the skin of a vessel are called her *Moulded Dimensions*, to distinguish them from her outside dimensions. The *breadth moulded* is the extreme breadth from inside to inside of the skin; the *moulded length* is measured from the after-edge of the rabbet of the stem to the forward edge of the rabbet of the stern-post; and the *moulded depth of immersion* amidships is measured from the plane of flotation down to the upper edge of the rabbet of the keel. By the *moulded displacement* is meant, the internal capacity of the skin of the vessel, below the plane of flotation.

When the naval architect makes his design represent the moulded figure and dimensions of the ship, it is necessary that he should consider in what proportions the real displacement, stability, and other quantities for the real external figure, will be greater than the corresponding quantities as computed for the moulded figure.

The *bends* or *wales*, being the part of the skin of the vessel at and near the load-water-line, depend for their thickness upon the material of the skin, and the strength which it is required to have. In wooden vessels, that thickness may be estimated as ranging from $\frac{1}{10}$ th to $\frac{1}{5}$ th of the *extreme half-breadth, moulded*; and in iron ships, at from $\frac{1}{10}$ th to $\frac{1}{5}$ th of the thickness proper for wooden ships of the same size. The thickness of the bends having been determined, let the proportion which it bears to the *mean half-breadth of the plane of flotation* (or extreme half-breadth \times coefficient of fineness of the load water-section) be denoted by *m*; also let the thickness of the skin of the floor be determined, and let

the proportion which it bears to the midship draught of water be denoted by n ; and let the fraction by which the real length on the load-water-line is to exceed the moulded length be denoted by p . Then the following proportions will be correct enough for a first approximation:—

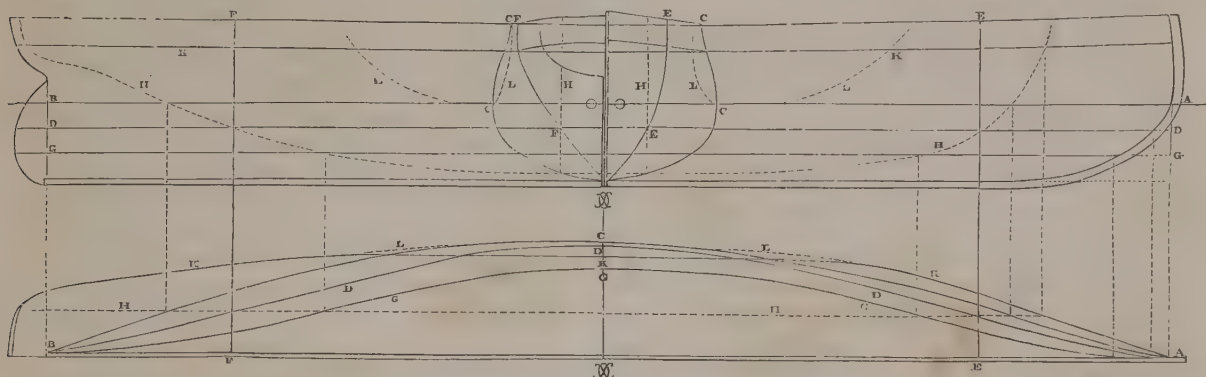
MOULDED : REAL	
Area of Plane of Flotation,.....	$1 : 1 + m + p$
Displacement,.....	$1 : 1 + m + n + p$
Surface Stability,.....	$1 : 1 + 3m + p$
Augmented Surface,.....	$1 : 1 + \frac{m+n}{2} + p$

194. *Midship Section*.—The upper diagram in Fig. 1 represents the sheer plan of a ship, in which the figure and dimensions of the longitudinal section have been laid down, and the length, A B, has been divided into fore-body and after-body (there being no middle-

body in the example chosen). The lower diagram represents the half-breadth plan, which at first contains only the length, A B, divided at \oplus into fore-body and after-body; a straight line drawn parallel to A B, at a distance representing the half-breadth (or “half-siding” as it is called) of the keel; and a third straight line, \oplus C, representing the extreme half-breadth of the ship.

The next step is usually to design the midship section, C C \oplus C C. The general character of this section below water is supposed to have been already so far determined, that its coefficient of fineness is known, at least approximately. In designing its figure in detail, the naval architect has to keep in view at once stiffness, steadiness, easy rolling, economy of power, and in sailing vessels, weatherliness. As regards the figure of the

Fig. 1.



midship section between wind and water, two main classes of vessels may be distinguished; those which are to float at a nearly constant draught of water (as ships of war, and yachts), and in which the best forms are unquestionably those which roll isochronously, as described in Article 126A; and those which are to float at a variety of different draughts (as merchant ships), and in which, in order that the stability may vary as little as possible, it is preferable to make the section between wind and water straight and vertical, or nearly so.

As regards the *floor*, or lowest part of the section, a sharp or rising floor is favourable to steadiness and to sailing; a flat floor is convenient for stowage, for the arrangement of engines, and for taking the ground, and is sometimes necessary when the draught of water is limited. The *bilges*, which unite the floor with the sides, are said to be “hard” or “easy” according as their curvature is more or less sharp. A hard bilge is considered to promote steadiness and weatherliness, so as to make up somewhat for flatness of the floor: but it is unfavourable to the fairness of the lines of the fore and after body: an easy bilge is on the whole the most favourable to good forms of water-line.

195. The *Leading Water-line* is that whose figure is first designed. It may be the load-water-line; but inasmuch as the water-line (D D in the figure) at one-third of the immersed depth, is approximately a mean of the other water-lines as to fineness, it is advisable to begin with it. The influence of the forms of water-lines upon resistance and speed, has been explained and exemplified in Articles 159 to 166 inclusive.

The fairness of this leading water-line is of primary importance to speed and economy of power. The variety of curves possessing the requisite quality of fairness is infinite; and the naval architect

may prefer to be guided wholly by his eye in designing this and other lines of the ship; but should he choose to avail himself of certain definite curves, he may be aided by some mechanical and geometrical processes which will be described in the second Division of this treatise.

196. *Balance Sections*.—The term “balance sections” is applied to a pair of vertical cross sections, one near each end of the vessel, which are designed after the midship section and leading water-line. Their position is optional; but in most cases it is convenient to place them (as directed by Fincham in his *Outlines of Naval Architecture*) at *one-third of the length of the fore-body and after-body respectively, from the ends of the line of flotation, A B*. They are so placed in the figure, where the forward balance section is marked E, and the after balance section, F. In vessels with very fine and sharp ends, it may sometimes be convenient to place the balance sections at the middle of the length of the fore-body and after-body respectively.

One half-breadth in each of those sections is already determined—viz., where they intersect the leading water-line, D D. The sections are completed according to the judgment of the naval architect; and their form is generally such as to give sharpness to the floor for resisting rolling and leeway, to make the water-lines gradually become finer from above downwards, and to flare out above water, for the sake of liveliness in pitching and scending, and of giving sufficient breadth to the decks, especially towards the stern of the vessel.

197. *Additional Water-lines* (such as A C B, and G G) can now be designed in any required number above as well as below water, by drawing (either by the eye, or by processes to be described in the Second Division) a series of fair curves through the points

where a series of horizontal planes cut the midship and balance sections.^o

198. *Buttock Lines.—Riband Lines.*—Additional longitudinal sections, or bow and buttock lines, can now be laid down, by first drawing straight lines (such as those marked H) on the half-breadth plan and body plan to represent the planes of those sections, and then marking on the sheer plan the points where those planes intersect the midship and balance sections, and the water-lines, and drawing fair curves through those points. An example of a longitudinal section is represented by the dotted line, marked H H in the sheer plan, Fig. 1.

The fairness of the buttock lines, or after-parts of the longitudinal sections, is so important (for reasons stated in Article 159), that the naval architect may sometimes find it advisable to design a *leading buttock-line* before any of the water-lines, and then adapt the water-lines of the after-body to it.

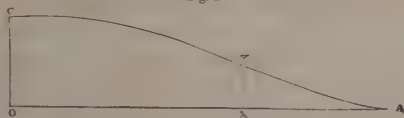
Riband or *Diagonal lines* are oblique longitudinal sections, sometimes used in testing the fairness of the body; but it is unnecessary to refer to them further here, as their construction will be described in the Second Division.

199. *Additional Cross-sections*, in any number that may be considered necessary, can now be constructed, so as to complete the body-plan, by taking their half-breadths at the several water-lines from the half-breadth plan. None of these are shown in Fig. 1; but numerous examples of them are contained in the Plates. In Article 91A it has been explained, how the figure of a cross-section may be modified without altering its area.

200. By the *Main-breadth-line* is meant, a line on the surface of the vessel, cutting each of the cross-sections at the point where its breadth is greatest. Its figure can be constructed on the three plans after the cross-sections have been completed. It is represented in Fig. 1 by the dotted curves marked L. It forms, of course, the outline of the half-breadth plan, in which it very often coincides with the load-water-line in the middle part of its length, and with the gunwale or the plank-sheer near the head and stern. In former times, one of the first steps taken in designing a ship was to assume a figure for the main-breadth-line; but,

* The following formulæ give good approximations to the relations between the coefficient of fineness of a really fair water-line, and its breadth at the midship and balance sections respectively.

Fig. 2.



In Fig. 2, let \overline{OC} , the extreme half-breadth of the water-line, CYA , be denoted by b ; let a denote the length \overline{OA} from the extreme half-breadth to the end; x , the distance \overline{OX} from the extreme breadth to the balance section; y , the half-breadth at the balance section; f , the coefficient of fineness;

Then make

$$c = \left(1 - \frac{x^2}{a^2} - \frac{y}{b}\right) \div \left(\frac{x^2}{a^2} - \frac{y^2}{b^2}\right);$$

and we have,

$$f = \frac{9-c}{12}; \quad c = 9 - 12f; \quad \frac{y}{b} = 1 - c \cdot \frac{x^2}{a^2} + (c-1) \frac{x^3}{a^3}.$$

When the curve is a parabola of the third or second order, these formulæ are exact: in other cases they are approximate.

For particular positions of the balance section, the formulæ take the following values:

$$\text{When } x = \frac{2}{3}a;$$

$$c = \frac{19b-27y}{4b}; \quad f = \frac{17b+27y}{48b}; \quad \frac{y}{b} = \frac{48f-17}{27} = \frac{19-4c}{27}.$$

$$\text{When } x = \frac{1}{2}a;$$

$$c = 7 - \frac{8y}{b}; \quad f = \frac{b+4y}{6b}; \quad \frac{y}{b} = \frac{6f-1}{4} = \frac{7-c}{8}.$$

as its shape has little direct influence on the qualities of the ship, that method is seldom followed now.

When a ship has a "straight of breadth" vertically; that is, when her cross-sections are partly vertical at the sides, there are two main-breadth-lines, at the upper and lower boundary of the straight of breadth respectively.

201. *Sheer-lines.—Head and Stern.*—The under side of the *gunwale*, marked K in the figure, is designed so as to form a fair curve on the half-breadth plan. Its form near the bow is like that of a water-line; but in most cases somewhat fuller. Near the stern, its figure is fuller still, with a view to giving a convenient breadth to the after part of the decks. It may sometimes be advisable to design the gunwale before completing the cross-sections.

In almost every vessel the gunwale has an upward curvature longitudinally, called the *sheer*. The true practical object of this is, to protect the vessel against waves breaking over her, by giving her greater height out of the water at the bow and stern, where her vertical motion relatively to the water is greatest.

The bow, as having more vertical motion relatively to the water than the stern, and also as being more exposed to the waves, requires the greater height of sheer. This may be effected by making the curvature of the sheer vanish, or nearly so, at the stern, and increase gradually in sharpness towards the bow. The following Table is deduced from the plans of various actual vessels:—

	Height of the Gunwale above its lowest point, in fractions of the length on the load-water-line.	
	At the Stern.	At the Head.
Great Sheer,.....	0.017	0.033
Medium Sheer,.....	0.01	0.02
Small Sheer,.....	0	0.01

The less lively a vessel is in pitching and scending, the more sheer does she require; and it is specially needed at the bow of a vessel with fine lines, and little or no flaring out above the water-line.

Large vessels have proportionally less sheer than small vessels, and ships of war than merchant-ships. The reason in the latter case is, that the decks follow the form of the sheer, and that decks with much sheer are inconvenient for working guns.

The rail above and parallel, or nearly parallel, to the gunwale, usually called the *rougntree rail*, has the same, or nearly the same figure.

The design of the *figure-head* or *cuthwater* (which projects like a sort of beak in front of the stem), and that of the stern, are matters to be regulated chiefly by the taste of the naval architect. In many vessels the figure-head is dispensed with (as in Fig. 1 of this chapter).†

202. *Use of Models in Designing Ships.*—A model, to be used by a naval architect in designing a ship, is usually composed of two sorts of soft wood of different colours, such as fir and cedar, in alternate layers, screwed, pinned, or glued together. The seams between the layers represent water-lines. The model usually represents the starboard half of the vessel, and has a plane side, representing the longitudinal midship plane, on which the sheer-plan is drawn. Its curved side is then gradually carved, shaved, and filed to such a form as to satisfy the eye and the judgment of

† The figure-head seems to have originated in the beak of the ancient galley, which projected straight-forward very near the water-line, was shod with iron, and was used as a weapon of attack. As that use of the beak became obsolete, it appears gradually to have crept upwards, and become convex; so that in the ships of the present time the head-rails, or side-pieces of the figure-head, form continuations of sheer-lines, and the knee of the head, or central beak-shaped part, has become unfit for its original purpose, and merely helps to support the bowsprit.

the designer; the touch also is used, by passing the hand over the model, as a test of the fairness of its figure.

The coefficient of fineness of a model may be determined by filling a trough with water exactly to the level of a suitable outlet, carefully lowering the model into the water until it is immersed exactly to the load-water-line, and finding, by measurement or by weighing, the volume of water which is made to run over. That volume will be equal to the displacement of the model; and being divided by the product of the principal dimensions of the model, the quotient will be the coefficient of fineness.

Another mode of roughly determining the displacement of a model is to separate the part below the load-water-line from the rest, weigh that part, and compare its weight with that of a rectangular block made up of the same materials in the same way; when the proportion of the volumes may be taken as approximately the same with that of the weights.

To find approximately the centre of buoyancy of a model, hang up the part below the load-water-line by a fine thread to a single pin in an exactly vertical board, so that the plane side of the model shall be in contact with the board. Mark the two points where a plumb-line hanging from the same pin passes the edges of the model, and draw a straight line on the plane side of the model through those two points. Repeat the experiment with the model hanging in a position as nearly as may be at right angles to its former position; the intersection of the two lines on its plane side will correspond to the centre of buoyancy of the vessel represented by the model. In models whose layers are screwed together, the accuracy of the result of this process may be inter-

fered with by unequal distribution of the screws; hence, for the purpose of finding the centre of buoyancy, the best fastening is made with wooden pins or glue.

Models are sometimes built up of small prismatic bars of wood, so as to show the figures of vertical and oblique as well as horizontal sections.

A model may be used either to construct drawings from, or directly in laying off the figures of the pieces of the ship on the mould-loft, without the intervention of drawings. Those processes will be further treated of in the Second Division.

203. *Summary of Calculations.*—It may here be useful to give a summary of the calculations which may be performed relatively to a design for a ship, together with references to the Articles in the preceding chapters of this Division, where the rules for those calculations are explained.

Subjects.	Chap.	Seet.	Articles.
Displacement and Centre of Buoyancy,.....	iii.	i.	86- 99
Approximate Transverse Stability and Metacentre,.....	iii.	ii.	100-106
(The Metacentre having been determined, the position where the Centre of Gravity ought to be is known.)			
Examples of those Calculations,.....	iii.	iii.	107-109
Exact Stability—Centres of Immersion and Emersion; Axis of Level Motion, &c.,.....	iii.	iv.	110-118
Longitudinal Stability; Centre of Plane of Flotation; Longitudinal Metacentre; Trim,.....	iii.	v.	119-128A
Time of Oscillation in Still Water; Properties with respect to Isochronism, and Easy and Uneasy Rolling,.....	iv.	i.	125-133
Properties with respect to Oscillation amongst Waves,.....	iv.	iii.	140-148
Augmented Surface; Relations between Speed and Power,.....	v.	iii.	161-172
Action of Propelling Apparatus,.....	v.	iv.	173-176
Power to carry Sail; Area of Sail, and Centre of Effort,.....	v.	v.	177-185
Action of Rudder; Proportions of Head-sail and After-sail,.....	v.	vi.	186-190

CORRECTIONS AND ADDENDA TO THE FIRST DIVISION.

Chapter IV., Section II., Article 134, page 68, column 1, line 6 from bottom, for "30 feet" read "43 feet."

Ibid., Column 2, line 4 from top, for "500 feet" read "560 feet."

Note to Chapter III., Section II.—*Centre of Gravity.*—It appears that the centres of gravity of the recently built iron-clad ships of war in the British Navy are in general about 2 feet below the plane of flotation, and 6 feet below the metacentre; being about 2 feet lower than the ordinary position of a ship's centre of gravity. (See a paper by E. J. Reed, Esq., Chief Constructor of the Royal Navy, in the Transactions of the Institution of Naval Architects for 1864.)

Note to Chapter IV., Section II.—*Waves.*—Much valuable information on the subject of Waves is contained in Mr. Thomas Stevenson's work on Harbours (Edinburgh, 1864).

ADDENDUM TO CHAPTER VI.—After a ship has been designed, and the sheer, body, and half-breadth plans all faired, it may sometimes be requisite to alter the form of the ship slightly, so as to obtain an increased or diminished displacement. This may often be effected by retaining the water-lines in the half-breadth plan, and increasing or diminishing the common intervals between the water-lines; or the transverse sections of the body plan may remain unaltered, but the interval between them be increased or diminished.

CASE I.—Let D represent the displacement according to the prepared plan, Δ the increase or decrease of the displacement, d the common interval between the

horizontal sections, and z the quantity by which this common interval is increased or diminished. Then

$$z = d \cdot \frac{\Delta}{D};$$

also, since the number of intervals will remain unaltered, the increase or decrease in the whole distance between the extreme horizontal sections will be

$$= \text{Original distance between extreme horizontal sections} \times \frac{\Delta}{D}.$$

Again, the distance of the centre of gravity of the displacement, bounded by the extreme longitudinal section, will be to the original distance as $D + \Delta : D$, or as $d + z : d$.

Also, the moment of inertia of the load-water-section remaining the same, the height of the metacentre above the centre of buoyancy will be to the original height as D to $D + \Delta$.

CASE II.—When the forms of the vertical sections are retained, but the interval between them is altered, let l represent the interval between the vertical sections, and y its increase or decrease, when the displacement is increased or decreased by the quantity, Δ . Then

$$y = l \times \frac{\Delta}{D}.$$

The depth of the centre of buoyancy below the load-water-line remains unaltered, as also the height of the metacentre above the centre of buoyancy.

DIVISION SECOND.

GEOMETRY OF SHIPBUILDING.

CHAPTER I.

SUMMARY OF GEOMETRICAL PRINCIPLES.

ARTICLE I. *Subjects of this Chapter stated.*—The present Chapter is designed to give a summary of certain general principles and methods in geometry which have frequently to be used during the operations of constructing the detailed plans of a ship, whether on paper or on the mould-loft floor. The first and smaller Section of the chapter relates to geometry in two dimensions only, and is devoted specially to the explanation of methods of drawing with precision continuous or "fair" plane curves, so as to pass through given points, and touch given lines—a problem of very frequent occurrence in naval architecture. The first two Articles of that section relate to the case in which the naval architect in designing some line of a ship, has not adopted any definite curve known in mathematics, but has fixed the positions of certain points in the line, or of certain straight lines which touch it, so as to satisfy his eye and judgment; and the problem is to draw a truly fair curve through those points, and touching those straight lines, by the simplest and quickest process; and such is the case that oftenest occurs in practice. The remainder of the Section relates to the drawing of some definite mathematical curves which have been already referred to, such as harmonic curves, trochoids, neoids, &c.

The second and larger Section contains a summary of the principles, so far as they are required in shipbuilding, of "descriptive geometry," or in other words, the mathematical theory of drawing; being the art of accurately representing figures of three dimensions on a plane surface.

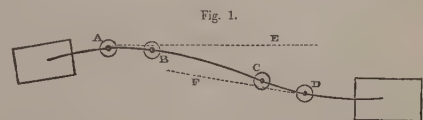
SECTION I.—CONSTRUCTION OF PLANE CURVES.

2. *Mechanical Construction by Springs and Battens.*—One of the easiest and commonest ways of drawing a fair curve through a series of points, is to fix a long, slender, elastic bar by means of pins, so that one of its edges shall traverse the given points, and to draw a curve along that edge. On paper, either a steel spring or a wood or whalebone batten is used for this purpose; on the mould-loft floor, a deal batten. The curve thus produced consists of a series of portions of elastic curves.

In order that a steel spring may give truly continuous curves when used in that way, it should fulfil two tests, viz.:—When simply laid on the paper in a free condition, it should be exactly straight; and when bent into a hoop, with its ends pinched

together, it should be exactly circular. The former of those tests only can be applied to a wooden batten.

When the given points are numerous and the curvature gentle the pins which guide the spring may be fixed in or alongside of the given points themselves. But when the given points are few and the curvature extensive, that mode of guiding the spring is apt to produce abrupt changes of curvature at or near the pins; and if very true continuity is required, the spring should be guided, not by pins at or near the given points, but by having its two ends fixed into blocks, as shown in Fig. 1; and it should then be



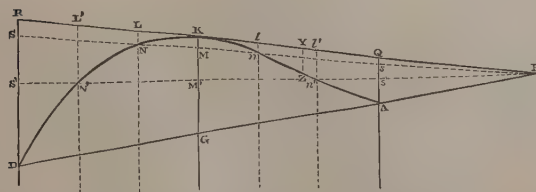
applied to not more than four given points at a time. The position of the spring is to be adjusted by shifting the two blocks about, until, as the case may be, it either traverses four given points, as A, B, C, D, or touches a given line (as A E) at a given point (as A), and at the same time traverses two other given points (as C, D); or touches two given lines (as A E and D F) in two given points (as A and D). The spring will then form a perfectly continuous curve.

3. *Interpolation of Points* is the name which may be given to the process of finding, either by geometrical construction or by calculation, intermediate points between a set of given points in a fair curve. It is capable of being carried to any degree of elaboration and complexity, by taking sets containing great numbers of given points; but inasmuch as for practical purposes it is desirable to avoid all complication beyond what is absolutely necessary, it may be laid down as a rule for the interpolation of points in the lines of a ship, that each set of data should be the same with those required in using a spring—viz., either four points, or three points and a tangent at one of them, or two points and the tangent at each of them. The curves constructed according to this rule are all made up of arcs of parabolas of the second and third orders; and consequently their areas are given *exactly* by Simpson's Rules—a property of some convenience in practice.

$$\overline{AE} = \frac{\overline{AC}^2}{\overline{AB}}.$$
$$\overline{A H} = \frac{\overline{A F}^2}{\overline{A B}}.$$

PROBLEM I.—(Fig. 3.)—Given, the ends, A, K, D, of three

Fig. 3.



Case First.—When the chord, DA , and the tangent, RKQ , are not parallel, find their point of intersection, P . If that point lies between D and A , it is itself a point in the curve.

$$\frac{\overline{KM}}{\overline{KG}} = \frac{\overline{KL}^2}{\overline{KR}^2}; \quad \frac{\overline{KM'}}{\overline{KG}} = \frac{\overline{KL'}^2}{\overline{KR}^2}; \quad \&c.;$$
$$\overline{MK} = \overline{MG} \cdot \frac{\overline{MN}^2}{\overline{MS}^2 - \overline{MN}^2};$$
$$\overline{SR} = \overline{DR} \cdot \frac{\overline{MN}^2}{\overline{MS}^2 - \overline{MN}^2};$$

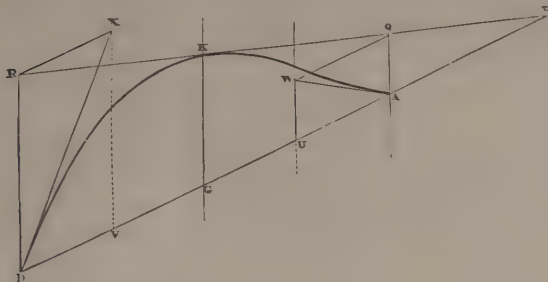
$$\overline{sQ} = \overline{AQ} \cdot \frac{\overline{MN^2}}{\overline{MS^2} - \overline{MN^2}};$$

(In the example shown in the drawing, $\overline{KL} = \frac{1}{8} \overline{KR}$; and accordingly we have $\overline{MK} = \frac{1}{8} \overline{GM}$, $\overline{SR} = \frac{1}{8} \overline{DS}$, and $sQ = \frac{1}{8} \overline{As}$.)

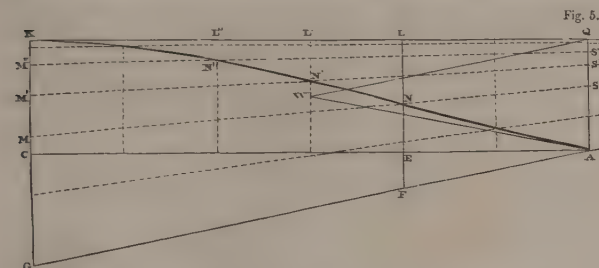
PROBLEM III.—Given, two points, and the tangents at those points, to interpolate intermediate points in the curve.

Method First:—By an intermediate tangent (see Fig. 4). (This is the easiest and most generally useful way of constructing

Fig. 4.



a fair curve, such as a water-line, so as to touch given straight lines at an indefinite number of detached points; both points and tangents being chosen at the discretion of the naval architect.)



being that at the entrance of the line. Draw AQ parallel to CK, cutting KQ in Q.

Bisect KQ in L', through which draw an ordinate parallel to KC, cutting the tangent AW in W. Join QW; and parallel to it draw AG, cutting KC (produced if necessary) in G, and KQ (produced if necessary) in P (unless W should happen to coincide with L', when AG will coincide with AC and be parallel to KQ).

Divide KQ into any convenient number of equal parts, by points such as L'', L', L, &c.; and through the points of division draw ordinates parallel to KC. Divide KG into the same number of unequal parts, by laying off from K distances proportional to the squares of the distances laid off on KQ; for example—

$$\overline{KM'} = \overline{KG} \cdot \frac{\overline{KL''^2}}{\overline{KQ^2}}; \text{ \&c.}$$

(In the figure, KQ is divided into six equal parts; and accordingly the distances laid off on \overline{KG} are respectively $\frac{1}{36}$, $\frac{4}{36}$, $\frac{9}{36}$, $\frac{16}{36}$, and $\frac{25}{36}$ of that line.)

Case First:—When KQ and GA intersect at a point, P, within a convenient distance, draw diverging lines from P to the points of division of KG; they will cut the ordinates corresponding to them in a series of points of the curve, such as N, &c.

Case Second:—When P is inconveniently far off, divide QA into parts proportional to those of \overline{KG} , and join the corresponding pairs of points of division, as M''S'', &c., by straight lines; these will cut the ordinates corresponding to them in the required points.

Case Third:—When G coincides with C, the lines through the points M'', &c., are all parallel to AC and KQ. (The curve KA is then a common parabola.)

Let A and D be the given points, and AW and DX the tangents to the curve at those points. The direction of the ordinates being fixed, divide the chord, AD, into four equal parts by the points U, G, V; and through A, U, G, V, D, draw five parallel ordinates. Let W and X be the points where the two given tangents cut the ordinates traversing U and V respectively. Through W draw WQ, and through X draw XR, both parallel to the chord AD, and cutting the endmost ordinates in Q and R respectively. Draw the straight line QR, cutting the middle ordinate in K; then K will be a point in the curve, and QKR a tangent at that point. Additional intermediate points may now be found as in Problem I.

Method Second:—(This method is specially suited to the drawing of a fair water-line with a given extreme breadth, and a given angle of entrance, and is therefore described as applied to that purpose.) In Fig. 5, let \overline{OK} represent the extreme half-breadth of the line to be drawn, and CA its length from the extreme half-breadth to the stem: then KQ parallel to CA is one of the two given tangents. Let AW be the other given tangent,

Fig. 5.

A Point of Contrary Flexure exists when KQ is not less than $\frac{1}{3}$ KP; which quantity is the longitudinal distance of that point from the extreme breadth. (In the figure, $\overline{KP} = 2\overline{KQ}$; and the point of contrary flexure consequently coincides with N, whose longitudinal distance from the extreme breadth is $\frac{2}{3}\overline{KQ}$.) The tangent at the point of contrary flexure cuts KQ at one-third of the longitudinal distance of that point from K; in other words, it cuts off one-ninth part from KP.

PROBLEM IV.—Given three points, K, N, A, and the tangent KQ; to find intermediate points. (Fig. 5.)

This problem is specially useful in constructing a fair water-line whose extreme half-breadth, \overline{OK} , and half-breadth at the balance-section, \overline{EN} , have been fixed by the naval architect.

In the ordinate at the balance-section, LNE (produced if necessary), lay off the distance—

$$\overline{LF} = \overline{LN} \cdot \frac{\overline{KQ^3}}{\overline{KL^3}}.$$

Draw the straight line AF cutting KC (produced if necessary) in G; then proceed as in Problem III., Method Second.

COROLLARY TO PROBLEM IV.—To find the tangent at A; through Q, parallel to AFG, draw QW, cutting the middle ordinate in W. The straight line WA will be the tangent required.^a

* The rules given in the above Article 3, are all based on elementary properties of parabolas of the third order.

Instead of finding interpolated points by geometrical construction, it may in certain cases be useful to find the lengths of interpolated ordinates by calculation. The following are the algebraical formulæ applicable to that purpose, for parabolic arcs of the third order.

CASE I.—Four points being given, let their abscissæ be denoted by x_1, x_2, x_3, x_4 , and their ordinates by y_1, y_2, y_3, y_4 .

Let x be any intermediate abscissa; then the corresponding intermediate ordinate, y , is given by the following formula:—

In conclusion of this Article, it may be remarked, that when the curvature is moderate, the parabolic curves of which it describes the geometrical construction do not differ to an extent material in practice from the elastic curves whose mechanical construction is described in Article 2.

4. Construction of Harmonic Curves.—The use of complete

$$y = y_1 \cdot \frac{(x-x_2) \cdot (x-x_3) \cdot (x-x_4)}{(x_1-x_2) \cdot (x_1-x_3) \cdot (x_1-x_4)} + y_2 \cdot \frac{(x-x_1) \cdot (x-x_3) \cdot (x-x_4)}{(x_2-x_1) \cdot (x_2-x_3) \cdot (x_2-x_4)} + y_3 \cdot \frac{(x-x_1) \cdot (x-x_2) \cdot (x-x_4)}{(x_3-x_1) \cdot (x_3-x_2) \cdot (x_3-x_4)} + y_4 \cdot \frac{(x-x_1) \cdot (x-x_2) \cdot (x-x_3)}{(x_4-x_1) \cdot (x_4-x_2) \cdot (x_4-x_3)} \quad (1.)$$

CASE II.—Three points and the tangent at one of them being given, let the abscissae of the points be x_1, x_2, x_3 , and their ordinates y_1, y_2, y_3 ; and let $\frac{dy}{dx}$ be the slope of the given tangent. Then—

$$y = \frac{dy_1}{dx_1} \cdot \frac{(x-x_2) \cdot (x-x_3) \cdot (x-x_4)}{(x_1-x_2) \cdot (x_1-x_3) \cdot (x_1-x_4)} + y_1 \cdot \frac{(x-x_2) \cdot (x-x_3) \cdot (x-x_4)}{(x_1-x_2) \cdot (x_1-x_3) \cdot (x_1-x_4)} \cdot \left(1 - \frac{x-x_1}{x_1-x_2} - \frac{x-x_1}{x_1-x_3} - \frac{x-x_1}{x_1-x_4}\right) + y_2 \cdot \frac{(x-x_1) \cdot (x-x_3) \cdot (x-x_4)}{(x_2-x_1) \cdot (x_2-x_3) \cdot (x_2-x_4)} + y_3 \cdot \frac{(x-x_1) \cdot (x-x_2) \cdot (x-x_4)}{(x_3-x_1) \cdot (x_3-x_2) \cdot (x_3-x_4)} + y_4 \cdot \frac{(x-x_1) \cdot (x-x_2) \cdot (x-x_3)}{(x_4-x_1) \cdot (x_4-x_2) \cdot (x_4-x_3)} \quad (2.)$$

CASE III.—Two points and the tangents at them being given, let x_1 and x_2 be the abscissae of the two points, and y_1 and y_2 their ordinates; and let $\frac{dy_1}{dx_1}$ and $\frac{dy_2}{dx_2}$ be the slopes of the two given tangents. Then—

$$y = \frac{dy_1}{dx_1} \cdot \frac{(x-x_1) \cdot (x-x_2)^2}{(x_1-x_2)^3} + \frac{dy_2}{dx_2} \cdot \frac{(x-x_2) \cdot (x-x_1)^2}{(x_2-x_1)^3} + y_1 \cdot \frac{(x-x_2)^2}{(x_1-x_2)^3} \cdot \left(1 - 2 \cdot \frac{x-x_1}{x_1-x_2}\right) + y_2 \cdot \frac{(x-x_1)^2}{(x_2-x_1)^3} \cdot \left(1 - 2 \cdot \frac{x-x_2}{x_2-x_1}\right) \quad (3.)$$

Problem IV. of the text may be considered as an example of Case II., in which—

$$y_1 \text{ denotes the extreme half-breadth;} \\ y_2, \text{ the half-breadth at the balance-section;} \\ y_3 = 0; \text{ and also} \\ x_1 = 0; \frac{dy_1}{dx_1} = 0; \text{ when Equation 2 becomes—} \\ y = y_1 \cdot \frac{(x-x_2) \cdot (x-x_3) \cdot (x-x_4)}{(x_1-x_2) \cdot (x_1-x_3) \cdot (x_1-x_4)} \cdot \left(1 + \frac{x}{x_3} + \frac{x}{x_4}\right) + y_2 \cdot \frac{x^2}{x_2^3} \cdot \frac{x-x_4}{x_2-x_4}$$

but this formula, which in its present shape is intricate and cumbrous, is capable of being more simply and conveniently expressed as follows. Let—

$$b \text{ denote the extreme half-breadth;} \\ a, \text{ the length from the ordinate } b \text{ to the end;} \\ x_2 \text{ and } y_2, \text{ the ordinate and abscissa at the balance-section;} \\ c, \text{ a coefficient, whose value (as in the formula given in the foot-note of Division I., Article 197), is—} \\ c = \frac{1 - \frac{x_2^3}{a^3} - \frac{y_2}{b}}{\frac{x_2^3}{a^3} - \frac{x_2^2}{a^2}} \quad (4.)$$

Then for any other abscissa x , the ordinate is—

$$y = b \cdot \left\{ 1 - c \cdot \frac{x^2}{a^2} + (c-1) \cdot \frac{x^3}{a^3} \right\} \quad (5.)$$

The area of the curve is—

$$\frac{9-c}{12} \cdot a \cdot b \quad (6.)$$

The slope of the tangent at the angle of entrance is given by the formula—

$$-\frac{dy}{dx} = (3-c) \cdot \frac{b}{a} \quad (7.)$$

Also let x' denote the abscissa of the point of contrary flexure; and

$$\frac{dy'}{dx'} \text{ the slope of the tangent at that point; then} \\ x' = \frac{ca}{3(c-1)} \quad (8.)$$

$$-\frac{dy'}{dx'} = \frac{c^2}{3(c-1)} \cdot \frac{b}{a} \quad (9.)$$

Reference must be made to the Transactions of the Royal Scottish Society of Arts for 1863, for Mr. Edward Sang's account of his method of computing ship's ordinates by means of what he calls "Safinet Equations," of the following form:—

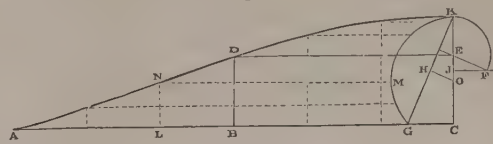
$$y = Y \cdot \left\{ \frac{1 + \frac{x}{a}}{1 + \frac{x}{r}} \right\}^s \cdot \left\{ \frac{1 - \frac{x}{b}}{1 - \frac{x}{s}} \right\}^t$$

in which

Y is the midship breadth of a given water-section;
 x is measured from the midship section, and is positive for the fore-body, and negative for the after-body;
 a denotes the length of the after-body upon the given water-section;
 b , the length of the fore-body upon the given water-section;
 s , an exponent on which the character of the bow depends;
 t , an exponent on which the character of the stern depends;
 r and s , coefficients which affect the form of the sides amidships.

harmonic curves for the water-lines of ships has been explained in Division I., Article 159. The naval architect, however, may prefer to make use of a greater or less portion of such a curve, according

Fig. 6.



to the general design of his vessel. For that purpose the following data are necessary and sufficient (see Fig. 6):—

The extreme half-breadth, \overline{CK} ;

The length, \overline{CA} ;

The half-breadth, \overline{BD} , at a balance-section midway between C and A.

This half-breadth may bear any proportion that the naval architect chooses to the extreme half-breadth, not less than $\frac{1}{2}$ nor greater than $\frac{3}{4}$. The former proportion gives a complete harmonic curve, or curve of versed-sines, whose coefficient of fineness is .5. The latter limit gives a common parabola, whose coefficient of fineness is $\frac{2}{3}$.

The construction is as follows:—Through D, parallel to AC, draw DE, cutting CK in E. Bisect CK in J, through which point draw JF parallel to AC. About E, with the radius EK, describe a circular arc, cutting JF in F, at the outer side of CK. Draw the straight line, FE, and produce it; and at right angles to it, draw KG.

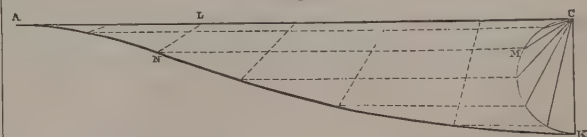
Bisect KG in H; and at that point erect a perpendicular to KG, cutting KC in O. About O describe the circular arc, KMG.

Divide the base, CA, into any convenient number of equal parts, and the arc, KG, into the same number of equal parts. Through each point of division of the arc (as M) draw a straight line (as MN). Through each point of division of the base (as L), draw an ordinate (as LN). The intersections of the ordinates with the corresponding lines parallel to the base will be points in the curve.

A straight line through O, parallel to CA, cuts the curve in its point of contrary flexure. The harmonic curve, when its breadth is moderate as compared with its length, approximates closely to the elastic curve, and to the cubic parabola.

5. Construction of Trochoids.—The principal properties and uses of the trochoid, or rolling-wave-line, have been stated in Division I., Articles 22, 44, 97, 135, 159, and 162. The easiest way of constructing it by finding a series of points is shown in Fig. 7,

Fig. 7.



where the curve is placed as if to form a buttock-line. The data are, the base \overline{CA} , and the greatest ordinate, \overline{CK} . On \overline{CK} as a diameter, describe a semicircle, and divide it into any convenient number of equal parts. Divide the base, \overline{CA} , into the same number of equal parts; then to each abscissa, such as CL, there will correspond a chord in the semicircle, such as CM. Complete the parallelogram formed by each abscissa and the corresponding

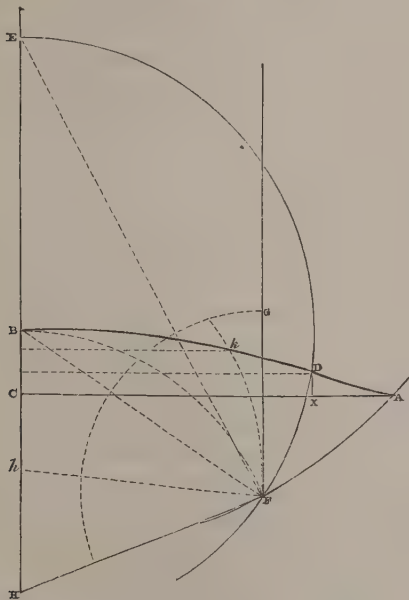
water-line which passes through or beyond P has only two points of minimum and one of maximum velocity of gliding, and does not raise supernumerary waves.

On the water-line, P Q, which traverses the point, P, itself, the velocity of gliding changes more gradually than on any other water-line having the same proportion of length to breadth. Water-lines possessing this character can be constructed with any proportion of length to breadth, from $\sqrt{3}$ (which gives an oval through L and P) to infinity. These are the lines called "lissoneoids" already mentioned in Division I., Art. 160, as being the fullest water-lines which do not raise supernumerary waves.

The construction of a whole series of neoids has been described, in order to make the principles more clear. For practical purposes, it can seldom be required to construct more than one such curve of a series at a time. Two cases may be distinguished, according as the proportions of the curve are to be arbitrarily chosen by the constructor, or fixed so as to give a lissoneoid.

CASE I.—To construct a Neoid curve of any given length, extreme half-breadth, and approximate fineness.

Fig. 8A.



In Fig. 8A, let \overline{CB} be the extreme half-breadth, and \overline{CA} the length. In \overline{CA} , take—

$$CX = CA \times \frac{2}{3} \text{ coefficient of fineness;}$$

and at X set up the ordinate, $XD = \frac{1}{3} \overline{CB}$.

About B, and through D, describe the circular arc, FDE, cutting CB produced in E. About E, through A, describe the circular arc, AFE, cutting the former circular arc in F. F will be the focus.

Through F draw FG parallel to BC; join FB, FE; and draw FH making the angle $BFH = BFG$, and cutting BC, produced if necessary, in H. Divide the angle, HFE ($= \frac{2}{3} BFG$) into a convenient number of equal parts, by lines diverging from F and cutting HE in a series of points (such as *h*). H, B, and E, are themselves three of those points. About the series of points thus found, describe circular arcs through the focus, F.

Divide CB into the same number of equal parts with the angle HFE, and through the points of division draw straight lines

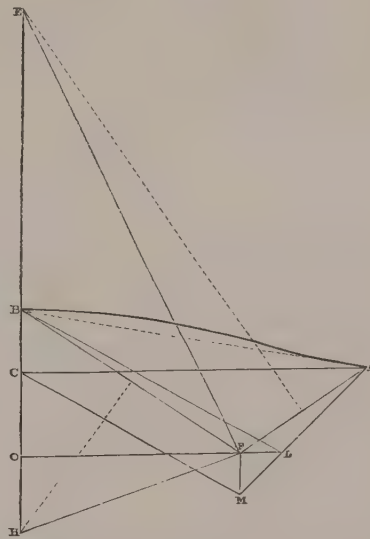
parallel to CA. The points, such as *k*, where those lines cut the arcs respectively corresponding to them, will be points in the required curve.

Sometimes, instead of fixing the approximate coefficient of fineness in the first instance, the position of the ordinate, $\overline{DX} = \frac{1}{3} \overline{CB}$ may be fixed (as when a fair water-line is to be drawn through a given point in a buttock-line whose distance from the axis, CA, is one-third of the extreme half-breadth, or a fair buttock-line through a given point in a water-line at one-third of the draught below the plane of flotation); and then the coefficient of fineness will be approximately—

$$\frac{5 \overline{CX}}{6 \overline{CA}}$$

CASE II.—To construct (approximately) a Lissoneoid of a given length and extreme half-breadth.—(Fig. 9.)—Let \overline{CB} be the

Fig. 9.



extreme half-breadth, and CA the length. Through C draw CM, making an angle of 30° with CA; through A draw AM, making an angle of 45° with AC, and cutting CM in M. Through B draw BL parallel to CM, cutting AM in L; through L draw LO parallel to AC; through M draw MF parallel to CB, and cutting LO in F. This point will be the focus. Then proceed to find centres, draw arcs, and intersect them by parallel lines, as in Case I. The two extreme centres, for the arcs through B and A respectively, are marked H and E, as in Fig. 8A.

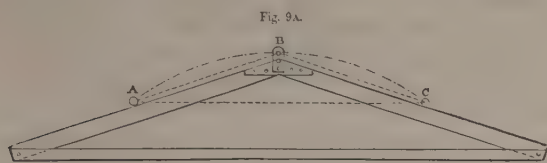
The straight line, BA, coincides almost exactly with a tangent to the curve at A; and the curvilinear area, ACB, is in ordinary cases nearly equal to the rectangle, $\overline{CB} \times \overline{OF} = \overline{CB} \times \overline{CA} \times$

$$\frac{\sqrt{3}}{1 + \sqrt{3}} = 0.634 \overline{CB} \cdot \overline{CA} \text{ nearly.}$$

6A. Construction of Circular Arcs of Large Radii.—The following well-known process is required in drawing an arc of a large circle in the mould-loft, when the centre of the arc is beyond the limits of the floor.

Let A, B, and C, be three points, through which a circular arc is to be drawn. Join BA, BC; then make a flat triangular mould, having two of its edges perfectly straight, and making with each other an angle equal to ABC, which may be tested by laying

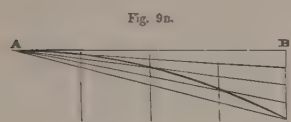
the mould on the floor. Each of those two edges should be of a length not less than the chord, AC . Fix a pair of truly cylindrical pins of equal diameter in A and C ; and fix a pencil to the angle



of the mould, so that when the two straight edges of the mould touch the pins at A and C , and are parallel to AB and BC , the point of the pencil shall exactly touch B . Then when the mould is moved so as to keep its edges touching the pins at A and C , the pencil will draw the required circular arc.

When two points only, as A and C , are given, together with a tangent at one of them, make the angle of the mould equal to the angle between that tangent and the chord, AC .

6B. *Construction of Common Parabolic Arcs.*—To draw an arc of a common parabola which shall pass through two given points, and touch a given line at one of those points, and whose axis shall lie in a given direction, the following is the process.



In Fig. 9B, let A and C be the two given points, AB the given tangent, and BC a line parallel to the given direction of the axis of the parabola, cutting the given tangent in B .

Divide AB into any convenient number of equal parts, and through the points of division draw lines parallel to BC . Divide BC into the same number of equal parts, and through the points of division draw straight lines diverging from A . The curve, AC , drawn diagonally through the network of straight lines thus formed, will be the required parabolic arc of the second order.

SECTION II.—ELEMENTARY RULES IN DESCRIPTIVE GEOMETRY.

7. *General Explanations.—Projection of Points and Lines.*—The whole art of drawing and laying off ships is a branch of descriptive geometry; by which is meant, the art of representing solid figures upon a plane surface. In the present Section are given some general elementary rules in that art, whose application is of very frequent occurrence in naval construction. The more special and complex rules will be given in the ensuing chapters, in treating of the particular parts of the construction of a ship to which those rules belong.*

By the *projection of a point* upon a given plane is meant, the foot of a perpendicular let fall from the point on the plane. For example, in Fig. 10, $XZZX$ represents a plane (called a *plane of projection*), A a point, and AB a perpendicular let fall from the point on the plane; the foot, B , of that perpendicular, is the projection of the point, A , on the plane, $XZZX$.

The position of a point is completely determined, when its

* For complete information on the subject of descriptive geometry, reference may be made to the works of Monge and Hachette in French, and of Dr. Woolley in English.

projections upon two planes not parallel to each other are known. In descriptive geometry, a pair of planes of projection at right angles to each other are used; and in general one of these is vertical and the other horizontal. Thus, in Fig. 10, $XZZX$ is the vertical plane of projection, and $XY YX$ the horizontal plane of projection; B is the vertical projection, and C the horizontal projection of the point, A ; and those two projections completely determine the position of the point, A ; for no other point can have the same pair of projections.

The *axis of projection* is the line, XX , in which the two planes of projection cut each other.

When the two projections of an object are shown in one drawing, it is convenient to represent to the mind that the following process has been performed:—Suppose that the vertical plane of projection is hinged to the horizontal plane at the axis, XX , and that after the projection of the object on the vertical plane has been made, that plane is turned about that axis until it lies flat in the position, $XZZX$, so as to be continuous with the horizontal plane: thus bringing down the projection, B , to b . This process is called the *rabatment* of the vertical plane upon the horizontal plane (to use a term borrowed from the French by Dr. Woolley). The two points, C and b , are in one straight line, perpendicular to XX . The process of rabatment may be conceived also to be performed upon a plane in any position, when a figure contained in that plane is shown in its true dimensions on one of the planes of projection.

The projection of a line is a line containing the projections of all the points of the projected line. The projection of a straight line perpendicular to the plane of projection, is a point; for example, the projection on the vertical plane, $XZZX$ (Fig. 10) of the straight line, AB , perpendicular to that plane, is the point, B . The projection of a straight line in any other position relatively to the plane of projection is a straight line. If the projected line is parallel to the plane of projection, its projection is parallel and equal to itself; thus the projection on the horizontal plane, $XY YX$, of the horizontal straight line, AB , is the parallel and equal line, CD . If the projected line is oblique to the plane of projection, the projection is shorter than the original line.

The projections on the same plane, of parallel and equal straight lines, are parallel and equal. The projections on the same plane, of parallel lines bearing given proportions to each other, are parallel lines bearing the same proportions to each other. When the plane of a plane curved line is perpendicular to a plane of projection, the projection of the curve on this plane is a straight line, being the intersection of the plane of the curve with the plane of projection. When the plane of the projected curve is parallel to a plane of projection, the projection of the curve on this plane is similar and equal to the original curve. In all other cases, it follows from the preservation of the proportions of a set of parallel ordinates amongst their projections, that the projections of a plane curve of a given algebraical order are curves of the same algebraical order. The projections of a circle are ellipses; the projections of a parabola of a given order are parabolas of the same order. The projections of a straight tangent to a plane curve are straight tangents to the projections of that curve. The projections of a point of contrary flexure in a plane curve are points of contrary flexure in its projections.

A third plane of projection, perpendicular to the first two, is often employed, not as being mathematically necessary, but as

being convenient for the representation of certain lines. Thus (as already explained in Articles 86 and 87 of the First Division), the sheer plan, half-breadth plan, and body plan of a ship, are projections of certain lines upon the ship's surface on three planes at right angles to each other. Any two of those projections are mathematically sufficient to show the whole dimensions and figure of the ship, and from any two the third can be constructed; but it is convenient for purposes of measurement and calculation to have the whole three projections.

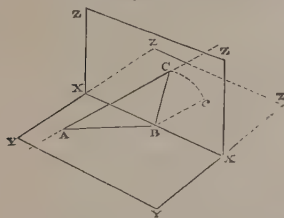
In the application of the rules about to be stated in the sequel of this Section, the two planes of projection may be held to represent any two of the three plans of a ship; and the axis of projection will then have the directions stated in the following Table:—

Plans represented by the Planes of Projection.	Direction of the Axis of Projection.
Sheer-Plan and Half-Breadth Plan,.....	Longitudinal.
Sheer-Plan and Body Plan,.....	Vertical.
Half-Breadth Plan and Body Plan,.....	Transverse.

Projections of figures upon planes oblique to the principal planes of projection may be used for special purposes.

8. *Traces of Lines and Surfaces.*—By a trace is meant, the intersection of a line with a surface, or of one surface with another.

Fig. 11.



The trace of a line upon a surface is a point; the trace of one surface upon another is a line.

The water-lines of a ship may be regarded as the traces of its surface upon a series of horizontal planes, and the cross sections as the traces of the

same surface upon a series of transverse vertical planes.

In descriptive geometry the term *traces* is specially employed, when not otherwise specified, to denote the intersections of a line or surface with the two planes of projection.

The position of a *straight line* is completely determined when its traces are known. For example, the straight line, A C, in Fig. 11, has its position completely determined by its traces, A and C, being the points where it cuts the two planes of projection. The *rabatment* of the trace, C, is represented by *c*.

A straight line parallel to one of the planes of projection has only one trace, being the point where it cuts the other plane of projection.

A straight line parallel to the axis of projection has no traces.

The traces of a *plane* are straight lines, which (unless they are both parallel to the axis of projection), meet that axis in one point. The position of a plane is completely determined when its traces are known. For example, the plane A B C in Fig. 11 has its position completely determined by its traces B A and B C.

A plane perpendicular to one of the planes of projection has its trace on the other plane of projection perpendicular to the axis of projection.

A plane perpendicular to both planes of projection has for its traces two lines perpendicular to the axis. Thus, in Fig. 10, Article 7, the traces of the plane, A B C D, are D C and D B, both perpendicular to X X. A plane parallel to one of the planes of projection has a trace on the other plane of projection only, being a straight line parallel to X X.

If a plane traverses a straight line, the traces of the plane traverse the traces of the line.

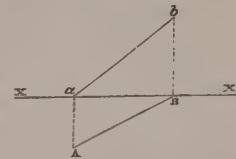
9. *Rules relating to Straight Lines.*—In each of the figures illustrating the following rules, the axis of projection is represented by X X; and in general, the part of the figure above that line represents the rabatment of the vertical plane of projection, and the part below, the horizontal plane of projection. The projections of points on the horizontal plane are in general marked with capital letters—the projections on the vertical plane, with small letters.

RULE I.—*Given* (in Fig. 12) *the traces, A, b, of a straight line, to draw its projections.* From A and b let fall A a and b B perpendicular to X X. Then a will be the vertical projection of the trace A, and B the horizontal projection of the trace b. Join a b, A B; these will be the projections required.

(It may here be remarked, that a A and a b are the traces of a plane traversing the given line, and perpendicular to the vertical plane of projection; and that B A and B b are the traces of a plane traversing the given line, and perpendicular to the horizontal plane of projection.)

RULE II.—*Given* (in Fig. 12) *the projections, A B, a b, of a straight line, to find its traces.* From a and B, where the given projections meet the axis, draw a A and B b perpendicular to X X, cutting the given projections in A and b respectively. These points will be the required traces.

Fig. 12.

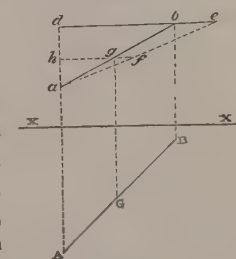


RULE III.—*Given the projections of two points, A, a, B, b* (Fig. 13), *to measure the distance between them.* Join a b, A B; these will be the projections of the straight line to be measured. Through either end of either of those projections (as b) draw d b e parallel to X X; through the other end, a, of the same projection, draw a d perpendicular to X X, cutting d b e in d; make d e = the other projection, A B; join a e; this will be the length required.

The same operation may be performed on the other plane of projection.

RULE IV.—*Given* (in Fig. 13) *the projections, A, a, of a point, and the projections, A B, a b, of a straight line through that point; to lay off a given distance from the given point along the given line.* In any convenient position, draw a straight line, B b, perpendicular to X X, meeting the given straight line in two points, B, b, which are the projections of one point; then perform the construction described in Rule III., so as to find a e. From the point, a, in the line, a e, lay off the given distance, a f. Through f draw f h parallel to X X, cutting a b in g; a g will be one of the projections of the given distance. Then draw g G perpendicular to X X, cutting A B in G; A G will be the other projection of the given distance.

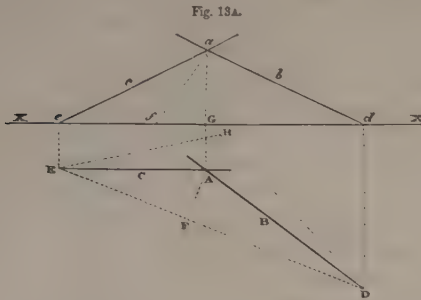
Fig. 13.



Another method of finding G is to lay off A G = h f.

RULE V.—*Given* (in Fig. 13) *the projections, a b, A B, of a straight line; to find the angle which it makes with one of the planes of projection* (for example, the horizontal plane). Perform the construction described in Rule III.; then d e a is the angle made by the given line with the horizontal plane. The same construction, performed in the horizontal plane of projection, will give the angle made by the given line with the vertical plane of projection.

RULE VI.—Given (in Fig. 13A) the projections, a, b and AB, ac and AC , of a pair of straight lines intersecting each other in the

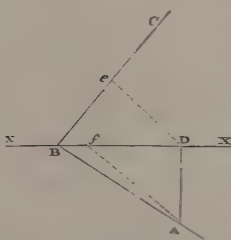


point whose projections are a, A ; to find the angle between those lines. In either of the planes of projection (for example, the vertical plane) find the points, d, e , where the projections of the given line cut the axis, XX ; these will be also the vertical projections of the horizontal traces of the lines. Through e and d draw eE, dD , perpendicular to XX , cutting AC and AB in E and D respectively; these points will be the horizontal traces of the lines. Join DE , and on it let fall the perpendicular, FA . Join Aa (which of course is perpendicular to XX); let it cut XX in G . Make $Gf = AF$, and join af . In FA produced, take $FH = af$; join HE, HD ; EHD will be the angle required.

10. Rules relating to Planes.—RULE VII. Given the projections of two lines in a plane; to draw its traces. Find, by Rule II., the traces of the lines; two straight lines drawn through those traces on the planes of projection will be the required traces of the plane.

RULE VIII.—Given (in Fig. 14) the traces of a plane, BA, BC ; to find the angle which it makes with one of the planes of projection (for example, the vertical plane). From any convenient point, A , in the horizontal trace, let fall AD perpendicular to XX . From D let fall De perpendicular to BC . In DB lay off $Df = De$. Join fA (this will represent the perpendicular distance from BC of the point whose projections

Fig. 14.

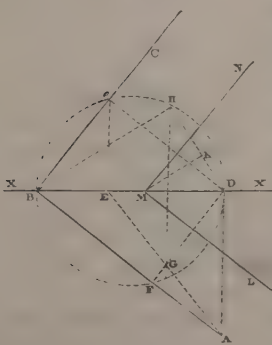


are D and A). AfD will be the angle required.

That angle is sometimes called the *bevel* of the given plane.

RULE IX.—Given (in Fig. 15) the traces of a plane, BA, BC ; to find the angle which it makes with the axis of projection, XX . In either of the two traces (for example, BA), take any convenient point, A , from which let fall AD perpendicular to XX ; and on BD as a diameter describe a circle. From D let fall perpendiculars, De, DF , on the two given traces. From the point, e , thus found on the opposite trace to that on which the point, A , was assumed, let fall eE perpendicular to XX ; join EA , cutting DF in G . From G draw GH perpendicular to XX , cutting the circle in H ; DBH will be the required angle.

Fig. 15.

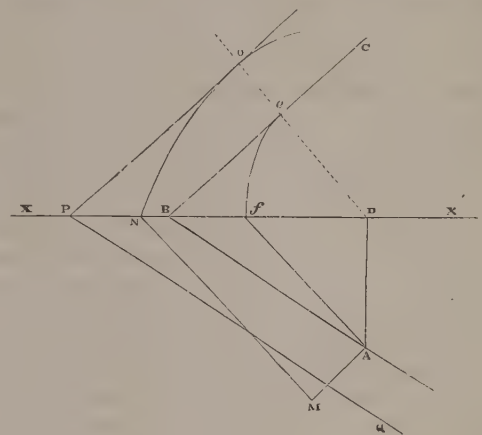


join EA , cutting DF in G . From G draw GH perpendicular to XX , cutting the circle in H ; DBH will be the required angle.

RULE X.—Given (in Fig. 15) the traces of a plane, BA, BC ; to draw the traces of another plane, which shall be parallel to the given plane, and at a given perpendicular distance from it in either direction. Complete the construction described in Rule IX. Join DH (this represents the perpendicular distance of the point, D , in the axis from the given plane); then from H , along HD (or along DH produced, according to the direction in which the new plane is to lie), lay off the given perpendicular distance between the planes, HK . From K draw KM parallel to HB , cutting XX in M . From M draw MN parallel to BC , and ML parallel to BA ; these will be the traces of the plane required.

Or otherwise:—Complete the construction described in Rule VIII. (see Fig. 15A). Af is the rabatment of the intersection of the given plane, with a plane, ADe , perpendicular to the vertical trace, BC . Through A draw AM perpendicular to Af , and make AM equal to the given distance between the planes; draw MN

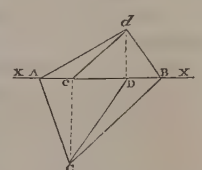
Fig. 15A.



parallel to Af , cutting XX in N . In De produced take DO , equal to DN . O is a point in the trace of the plane required. Through O draw OP parallel to BC , cutting XX in P ; and through P draw PQ parallel to BA . OPQ is the plane required.

RULE XI.—Given (in Fig. 16) the traces of two planes, $CA d$ and $CB d$; to draw the projections of the line of intersection of those planes. The traces of the required line are C and d , where the traces of the given planes intersect. From those points respectively, let fall Cc and dD , perpendicular to XX ; join CD, cd ; these will be the projections required.

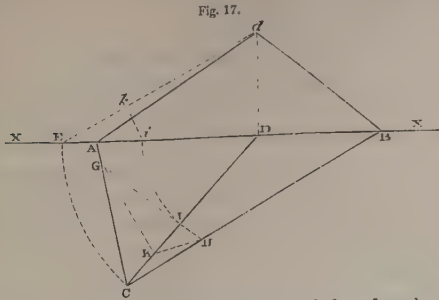
Fig. 16.



To find the projections of the point where a straight line intersects a plane (the traces of the line and of the plane being given), it is only necessary to draw the traces of two planes traversing the given line in convenient directions, and find the projections of the lines in which those two planes cut the given plane; the intersections of those projections will be the projections of the point required.

RULE XII.—Given (in Fig. 17) the traces of two planes, $C d A$, $CB d$; to find the angle between them. From either of the intersections of the traces (say d) let fall dD perpendicular to XX ; draw DC , joining D with the other intersection of the traces. Through any convenient point, I , in DC , draw GIH

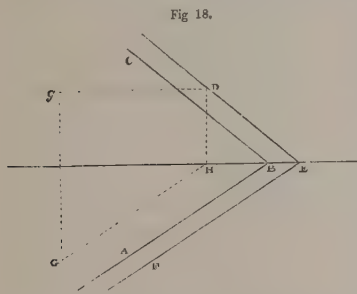
perpendicular to DC, cutting AC in G and BC in H. Along XX, lay off DE = DC, and Di = DI; join dE (this will



be the length of the line of intersection of the planes). From i let fall ik perpendicular to dE ; in IC , take $IK = ik$; join KG , KH ; GKH will be the angle required.

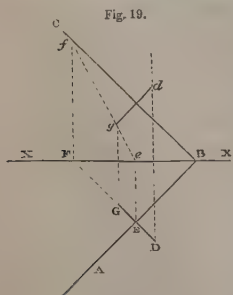
When the traces of the two given planes are inconveniently placed for the completion of the figure, we may substitute for either pair of traces, another pair of traces parallel to them, and more conveniently placed.

RULE XIII.—Given (in Fig. 18) the traces of a plane, ABC , and the projections of a point, G, g ; to draw the traces of a plane traversing the given point,



ing the given point, and parallel to the given plane. Through either of the projections of the given point (say G) draw GH parallel to the corresponding trace of the given plane, and cutting XX in H . (This will be one of the projections of a line through the given point, parallel to the trace, AB , of the given plane). Through H draw HD perpendicular to XX ; and through g draw gD parallel to XX , cutting HD in D (gD will be the projection and D one of the traces, of the line before-mentioned). Through D draw DE parallel to CB , cutting XX in E ; and through E draw EF parallel to BA : DEF will be the traces of the required plane.

RULE XIV.—Given the traces of a plane, EF, ED (in Fig. 18), and one projection of a point in that plane; to find the other projection of that point. Suppose g , the vertical projection of the point, to be given. Draw gD parallel to XX , cutting ED in D . From D let fall DH perpendicular to XX . From g draw gG perpendicular to XX , and from H draw HG parallel to EF ; the intersection of those lines, G , will be the required horizontal projection of the given point.

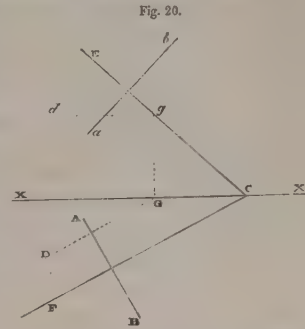


RULE XV.—Given (in Fig. 19) the traces, ABC , of a plane, and the projections, D, d , of a point; to draw the projections of a perpendicular let fall from the given point on the given plane. From one of the projections of the given point (say D) draw DEF perpendicular to the corresponding trace, BA , of the given plane, and cutting BA in E , and XX in F . From E let fall Ee perpendicular to XX ;

from F draw Ff perpendicular to XX , cutting the trace BC in f ; join fe ; from d draw dg perpendicular to BC , cutting fe in g ; and from g draw gG perpendicular to XX , cutting DF in G . DG and dg will be the projections of the perpendicular required.

RULE XVI.—Given (in Fig. 20) the projections of a point, D, d , and those of a line, AB, ab ;

to draw the traces of a plane which shall traverse the point, and be perpendicular to the line. Through one of the projections of the given point (say D), draw DG perpendicular to AB (the corresponding projection of the given line) cutting XX in G . Through G draw Gg perpendicular to XX ; through d , the other projection of the point, draw dg parallel to XX , cutting Gg in g ; through g draw EC perpendicular to XX in C ; and through C draw CF perpendicular to AB . ECF will be the traces of the required plane.



To draw the projections of a perpendicular to a given straight line in a given plane, find, by the preceding rule, the traces of a plane perpendicular to the given line; and then, by Rule XI., draw the projections of the line in which that plane intersects the given plane.

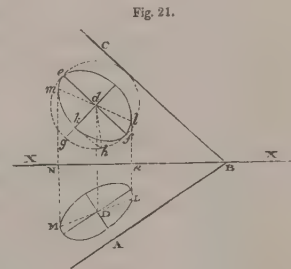
11. Projections of a Circle.—When an instrument which draws ellipses accurately is at hand, it may be used for the purpose of drawing the projections of a circle of a given radius, described about a given point in a given plane, and may thus facilitate much the solution of various problems. The following is the process for obtaining the projections of a circle:—

RULE XVII.—Given (in Fig. 21) the traces of a plane, ABC , and the projections of a point in that plane, D, d ; to draw the projections of a circle of a given radius, described in the given plane and about the given point.

For the vertical projection, describe about d a circle of the given radius, $df = de$, and draw the diameter, ef , parallel to the trace, CB ; ef will itself be the vertical projection of one diameter of the circle. Draw dg perpendicular to ef . Find, by Rule VIII., the angle which the given plane makes with the vertical plane of projection, and lay off gdk equal to the angle so found. From h , in the circle, draw hk parallel to fe , and cutting dg in k ; then dk will be the vertical projection of a radius of the circle perpendicular to ef . Then on the major axis, ef , and minor semi-axis, dk , describe an ellipse; that ellipse will be the required vertical projection of the circle.

The horizontal projection is obtained by a precisely similar process, Rule VIII. being now used to find the angle which the given plane makes with the horizontal plane of projection.

The two ellipses are both touched by a pair of tangents, Mm, Ll , perpendicular to XX ; and the diameters, lm, lM , are the projections of one diameter of the circle. The perpendicular distance, Nn , between those tangents, is equal to the diameter of the circle multiplied by the cosine of the angle which the given plane makes with XX , and is bisected by the line, Dd .



12. *Projections of Plane Curves—Interpolation of Points.*—It has already been stated in Article 7 that the projections of tangents to a curve are tangents to the projections of the curve; that the projections of parallel lines (such as ordinates), bearing given proportions to each other, are parallel lines bearing the same proportions to each other; and consequently, that the projections of a parabola of a given order are parabolas of the same order. Hence follows—

RULE XVIII.—*Given, for a plane curve, the projections of four ordinates, or three ordinates and a tangent at the end of one of them, or two ordinates and the tangents at their ends; to find the projections of interpolated points in that curve.* Interpolate, by the Rules of Article 3 of this Division, points upon equal numbers of equidistant ordinates on the two planes of projection; those points will be the required projections of interpolated points in the curve.

RULE XIX.—*Given one projection of a plane curve and the traces of its plane; to draw the other projection.* Take a sufficient number of points in the given projection; find the other projections of those points by Rule XIV., and draw a fair curve through them.

RULE XX.—*Given the projections of the base line (or axis of abscissæ), and of a series of parallel ordinates of a plane curve; to construct the curve itself* (in other words, to “rabat” the curve on the plane of the drawing). Find, by Rule VI., the angle between the base-line and the ordinates, and by Rule III. the lengths of the ordinates and of the several intervals of the base-line. From these data construct the rabatment of the base-line and ordinates, and draw a fair curve through the heads of the ordinates.

13. *Projected Plane Areas.*—RULE XXI.—*Given one of the projections of a plane figure, and the two traces of its plane; to find the area of the figure.* Measure and calculate the area of the given projection by Simpson's Rule. Find, by Rule VIII. of this Section, the angle made by the plane of the figure with the plane of the given projection; multiply that area by the secant of that angle: the product will be the area required.

(In Fig. 14, the value of the secant of the angle in question is given by the ratio, $\frac{FA}{FD}$.)

RULE XXII.—*Given the projections of a plane figure on three planes perpendicular to each other; to find its area.* Measure the areas of the three projections by Simpson's Rule; the square root of the sum of the squares of those three areas will be the area required.

If the plane of the figure is perpendicular to one of the three planes of projection, its projection on that plane will be a straight line, which has no area; so that the areas of the other two projections only will have to be measured.

If the plane of the figure is perpendicular to two of the planes of projection, and parallel to the third, its area is simply equal to that of its projection on the plane to which it is parallel.

14. *Curved Surfaces* (such as that of a ship) are represented in a drawing by means of the projections of certain curves contained in them; and those curves are usually sections of the curved

surface by certain planes (for example, water-sections, cross-sections, longitudinal sections, diagonal sections, &c.) Those planes of section are very often parallel to planes of projection (as in the case of longitudinal sections and cross-sections, and of water-sections of a ship on an even keel); sometimes they are oblique, as in the case of diagonal sections, water-sections not parallel to the keel, and other sections which will be more fully explained in the sequel. The solution of problems relating to curved surfaces in shipbuilding depends mainly on the principle, that at a given point in a continuous curved surface, all the tangent straight lines to curves in the surface traversing the given point, are contained in the tangent plane to the surface at the given point.

Hence follows—

RULE XXIII.—*Given the projections of the tangents to two curves in a curved surface at their point of intersection; to draw the traces of the tangent plane at that point.* Find, by Rule VII., the traces of a plane traversing the two given tangent lines.

When a tangent line at a point in a curved surface is parallel to a plane of projection, the trace of the tangent plane in that plane of projection is parallel to the trace of the tangent line; and such is generally the case in the plans of ships.

RULE XXIV.—*Given the projections of a point in a curved surface and the traces of a tangent plane at that point; to draw the projections of a tangent line at the same point, parallel to another plane whose traces are given.* Draw, by Rule XI., the projections of the line of intersection of the two planes; then draw, parallel to those projections, two lines through the corresponding projections of the given point of contact.

15. By *Development* is meant the process of drawing the figures which given lines on a curved surface would assume, if that surface were a flexible sheet, and were spread out flat upon a plane, without alteration of area, and without distortion. A limited class of curved surfaces only, known as *developable surfaces*, are capable of being so flattened. All lines, straight and curved, on a developable surface, continue to be of their true lengths, and to intersect each other at their true angles, after development. If two surfaces intersect each other in a curve of double curvature, and one of those surfaces is developable, the curve can be developed, and represented as a plane curve, by developing the developable surface.

Expansion is a process applied to surfaces which are not truly developable. It consists in covering the surface with a network, consisting of two sets of curves, which cross each other so as to form four-sided meshes; then conceiving the sides of those meshes to be inextensible strings, and drawing the network as it would appear if spread flat on a plane. By this operation, the meshes are both distorted and altered in area; the curves forming the network preserve their true lengths, but not their true angles of intersection; and all other lines on the surface are altered both in length and in relative angular position.

Examples of the use of development and of expansion will be given further on.

CHAPTER II.

ON THE BUILDING-DRAUGHT OF A SHIP.

SECTION I.—RELATIONS BETWEEN THE BUILDING-DRAUGHT AND THE CALCULATION-DRAUGHT.

16. *General Explanations.*—In Division I., Article 193, it has already been explained, that while the calculation-draught of a ship shows the figure of her external surface, the building-draught shows the figure of the internal surface of her skin and external surface of her framework (excepting the stem, keel, and stern-post, which in most vessels project beyond the external surface of the skin). Those two surfaces are exactly parallel to each other where the thickness of the skin is uniform, and nearly parallel when that thickness is variable.

The object of the present Section is to show how, when horizontal and vertical sections of the outer surface of the ship's skin are given, together with the thickness of the skin, the corresponding sections of the inner surface are to be constructed; in other words, the calculation-draught being given, how to draw the building-draught.

The same processes, with very slight modification, serve to perform the inverse operation of drawing the lines of the calculation-draught, when those of the building-draught are given.

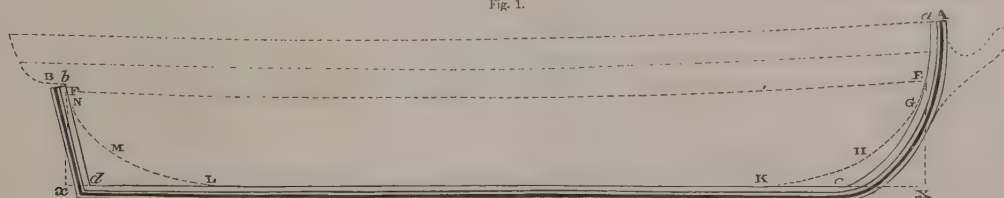
The *boundary line* of the outer surface of the skin of the ship, as shown on the calculation sheer-plan, is the *outside edge of the rabbet*, marked in Fig. 1 by a thick black line, which commences at the head of the stem (A), runs down the stem, along the keel, and up the stern-post, and terminates at the head of the stern-post (B).

That line may be regarded as the *trace of the outer surface* of the ship's skin upon a surface forming the side of the keel, stem, and stern-post, which surface is sometimes parallel to the midship longitudinal plane of the vessel.

The *trace of the inner surface of the ship's skin upon the same surface* is a longitudinal section (sometimes a bow and buttock line) on that surface, called the "*bearding-line*" (*a G H K L M N b*, Fig. 1). Its upper and forward part, *a G*, runs along the after edge of the rabbet of the stem, *a c*; its upper and after part *N b*, runs along the fore edge of the rabbet of the stern-post *d b*: its lower and middle part, *K L*, runs along the upper edge of the rabbet of the keel, *c d*. The dotted parts of the bearding-line, *G H K*, *L M N*, are called *stepping-lines*.

The *boundary of the inner surface* of the ship's skin is the

Fig. 1.



The latter operation, indeed, is somewhat the easier of the two; but the method of designing the calculation-draught first (as has been already stated in Division I., Article 193), is the more accurate, especially for wooden-skinned ships.

Throughout the present Section it will be assumed, that the planes of the horizontal and transverse sections in the building-draught and calculation-draught coincide with each other in position. In a later Section it will be explained, how to treat those cases in which the so-called horizontal sections or "level lines" in the building-draught are not truly horizontal, but parallel to a keel which is to be more deeply immersed at one end than at the other; while the transverse sections are to be not truly vertical, but perpendicular to the same sloping keel.

The details of the processes described in this Section are given as if for a wooden ship; because for such a ship, owing to the greater thickness of the skin and complexity of the framing, those processes are more difficult and tedious than for an iron ship; so that a student who has learned to perform them for a wooden ship, can perform them for an iron ship with comparative ease. For an iron ship, indeed, some of them become unnecessary.

middle of the rabbet; that is, the bottom of the triangular groove which (in wooden-skinned vessels) runs along the stem, keel, and stern-post. It is marked in Fig. 1 by a line close within and finer than the thick line which marks the outside edge of the rabbet.

The upper edge of the rabbet of the keel, produced if necessary, (*X c d x*), is the axis of projection of the sheer-plan on the building-draught. From the points E and F, where the sheer-line of the lowest deck that is permanently above water cuts the after edge of the rabbet of the stem and the forward edge of the rabbet of the stern-post respectively, two perpendiculars, EX and Fx, are commonly let fall upon that axis; these are called respectively the *foremost* and *aftermost perpendicular*; and the distance between them along the keel, Xx, is called "*the length of the ship between the perpendiculars.*"

The geometrical operations now to be described, for drawing the building-draught when the calculation-draught is given, consist mainly of three kinds of processes, viz.:—

I. Finding ordinates or half-breadths of the inner surface of the skin:

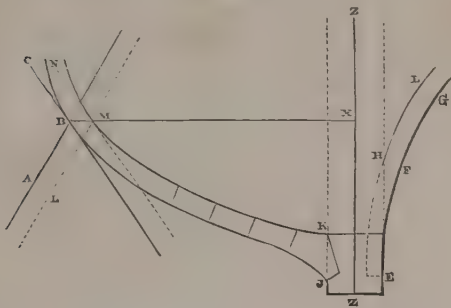
II. Finding points in, and constructing, the projection of the bearding-line, as already defined :

III. Finding points in, and constructing, the projection of the middle-rabbit line or boundary of the inner surface of the skin, on the sheer-plan.

In iron-skinned ships, the third of those operations may be regarded as unnecessary; and the second may in most cases be sufficiently well performed by taking the bearding-line as simply running down the after edge of the stem, along the upper edge of the keel, and up the forward edge of the stern-post, at a distance sideways from the midship longitudinal plane equal to the "half-siding" (that is, half the horizontal thickness) of the stem, keel, and stern-post.

17. *Half-breadths or Horizontal Ordinates in the Building-draught are found as follows:—*

Fig. 2.



Let Fig. 2 represent part of the body-plan of a ship, in which ZZ is the trace of the midship longitudinal plane, J the lower and K the upper edge of the rabbet of the keel, XB a horizontal ordinate, and the curve JB part of a cross-section of the outside surface of the skin of the vessel, as shown in the calculation-draught. It is required to find the point on the ordinate, XB, where the inside surface cuts it, the thickness of the skin being given.

On the midship section, and on every cross-section of the middle-body (if the ship has one) this problem is solved simply by laying off the thickness of the skin perpendicularly inwards from the curve, JB, at several points, and drawing through them a curve, which will be the midship cross-section on the building-draught, and will cut the ordinate, XB, in the point required.

On other cross-sections the method to be used is the following.

On the *half-breadth* plan of the calculation-draught, measure the angle which the water-line traversing the point B makes with the ordinate, XB; and on the body-plan lay off an equal angle, XBA. Draw a tangent, CB, to the cross-section at B.

Conceive that the ordinate, XB, is an axis of projection; and that the vertical plane and water-line plane traversing that ordinate are planes of projection; then AB and BC will represent the traces on those planes of a tangent-plane to the vessel's surface at B.

Then by Rule X. of the preceding Section, draw LM, MN, the traces of a plane parallel to the given tangent-plane, at a perpendicular distance equal to the given thickness of the skin; M will be the required point, where the inside surface of the skin cuts the ordinate; and XM, the ordinate, or half-breadth, of that surface at that point: which ordinate is to be laid down on the half-breadth plan also.

In the same manner any number of half-breadths in the building-draught may be found; after which moulded cross-sections (such as KM) and water-lines are to be drawn through their ends by suitable methods, such as those described in Chapter I., Section I., of this Division.*

18. *Construction of Stepping-lines.*—The bearding-line begins to rise above the upper edge of the keel, and becomes a stepping-line, towards each end of the vessel, at a pair of points (such as K and L in Fig. 1, Art. 16) where the rise of the floor begins to be so steep that the required thickness of skin cannot be obtained except by making its inner surface pass above the plane of the upper edge of the keel. Two cases may be distinguished, according as the rabbet of the keel is horizontal or inclined.

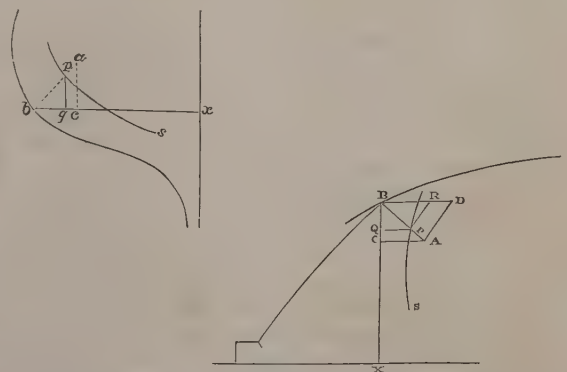
CASE I.—(*Rabbet Horizontal.*)—Let the right-hand side of Fig. 2 represent part of the body plan of a ship, E the lower edge of the rabbet of the keel, and EFG part of a steeply-rising cross-section. Through E draw the vertical straight line, EH, to represent the projection of the vertical longitudinal plane of the side of the keel.

Lay off a line, LH, straight or curved as the case may be, at a perpendicular distance inside of GFE, equal to the given thickness of the skin. (This may be done simply by taking that thickness in the compasses as a radius, describing a number of small arcs about points in GFE, and sketching a line to touch those arcs.) Then H, where that line cuts the vertical line EH, will be the transverse projection of a point in the stepping-line, and also the lower end of a cross-section in the building-draught. By laying off the height, EH, in its proper place on the sheer-plan, a point in the longitudinal projection of the stepping-line is obtained; and through a sufficient number of such points that line is traced.

* It is to be remarked, that strictly speaking, the process above described is approximative only; and the closeness of the approximation depends on the smallness of the thickness of the skin of the ship compared with the radii of curvature of the cross-section and water-line at B. To find with absolute exactness points on the inner surface of the skin, the following method may be used:—

Let BX and δx (Fig. 2A) represent the same ordinate in the half-breadth and body plans of a ship respectively. Through the points B and δ draw BA and δa , respectively normal to the curves at B and

Fig. 2A.



δ . These are the projections of a line normal to the surface at B, δ . In XB set off any distance, XC, and make αc equal to XC. Through C and c draw perpendiculars to XC and αc , meeting BA and δa in A and a respectively. Then A and a are the projections of a point in the normal to the ship's surface at B. Through A draw AD perpendicular to BA; make AD equal to αc , and join BD; then BD is the normal to the surface, rabbed on the horizontal plane; make BR equal to the thickness of the planking; draw RP perpendicular to BA; and P is the horizontal projection of a point in the surface of the ship. Draw PQ perpendicular to BX, cutting it in Q; in αb take αg , equal to XQ, and draw gp perpendicular to δx , cutting δa in p; then P and p are, respectively, the horizontal and vertical projections of a point in the inner surface of the ship. This point will not generally be situated either in the horizontal plane or the vertical plane through BX; it will therefore be necessary to find similar points corresponding to several of the level lines at the transverse sections. Curves PS and ps, drawn through the horizontal and vertical projections of these points, will give the projections of a line in the inner surface of the ship, and the trace of this line in the plane of each level line will give a point through which the level line, corresponding to the inner surface of the skin, will pass.

It is to be observed, that the line HL is a correct cross-section for a short distance above H only; and that the remainder of the cross-section which rises from H is to be drawn by finding ordinates, as in the preceding Article (17).

CASE II.—(When the rabbet of the keel has a slope).—Let

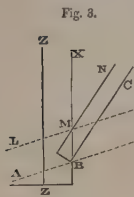


Fig. 3 represent part of the body-plan: ZZ , the projection and trace of the longitudinal midship plane; B , the lower edge of the rabbet of the keel, at a given cross-section on the calculation-draught, of which BC is the lower end; BX , the trace, upon that cross-section, of the vertical plane of the side of the keel.

Through B draw AB with a slope equal to that of the rabbet of the keel. Conceive BX to be a vertical axis of projection, where a longitudinal and a transverse plane of projection cut each other; then AB and BC will represent the traces of a tangent-plane to the ship's bottom at B . By Rule X. of the preceding Section, draw LM , MN , being the traces of a plane parallel to ABC and at a perpendicular distance from it equal to the given thickness of the skin: M will be the transverse projection of the required point in the stepping-line; and by transferring the height BM to the sheer-plan, the longitudinal projection of the same point may be found.

MN is the lower end of a cross-section on the construction-draught, subject to the remark already made at the end of Case I. The point, M , may be found by the method described in the note above.

19. The *Inner Edges of the Rabbets of the Stem and Stern-post* remain to be found, in order to complete the bearding-line at its ends.

METHOD I.—*Approximate process*, where the sides of the stem and stern-post are vertical, or nearly so.

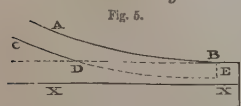
In Fig. 4, let XB represent part of a water-line, on the sheer-plan; EF , part of the front of the stem; GBH , part of the forward edge of the rabbet of the stem, cutting the water-line in B .

Find, from the half-breadth plan, the angle of obliquity of the water-line at B ; and lay off XBA equal to that angle. Conceive XB to be

an axis of projection, where the plane of the side of the stem (supposed vertical) and the horizontal water-section through B cut each other. Then AB and BC will represent the traces, on those two planes, of a tangent plane to the ship's surface at B . Draw, by Rule X. of the preceding Section, the traces LM and MN of a plane parallel to ABC , at a perpendicular distance equal to the thickness of the vessel's skin: then M will be a point in the bearding-line, or inner edge of the rabbet.

It is easy to see how the same process may be applied to the rabbet of the stern-post.

20. The *Bearding-line at fine hollow Water-lines* may be found by the following process:—Let Fig. 5



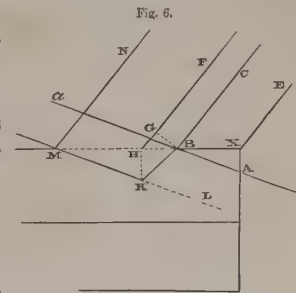
represent part of the half-breadth plan at and near the stem; XX , part of the longitudinal axis of the vessel; BD , the projection of the side of the stem; BA , the end of a fine hollow water-line on the calculation-draught.

Lay off, as in Case II. of Article 18, a line, EDC , at a distance from BA equal to the thickness of the skin of the vessel; D , where that line cuts BD , will be the horizontal projection of a point in the bearding-line; and the vertical longitudinal projection of the same point will be found by laying off, on the given water-line in the sheer-plan, the distance BD from the fore edge of the rabbet of the stem.

It is easy to see how the same process may be applied to the after-ends of water-lines.

21. *Projections of the Middle Rabbet.—Approximate Process.*—

Let Fig. 6 represent part of the sheer-plan: XBM , the projection upon it of a water-line; XE , part of the front of the stem; ABC , LMN , the traces of tangent planes to the outer and inner surfaces of the skin, at B and M respectively, found as in Article 19, in which the plane of the side of the stem is supposed to be vertical. It is



required to find the projection on the sheer-plan of the point where the middle rabbet cuts the given water-section, and the trace of the middle rabbet on the plane of that water-section.

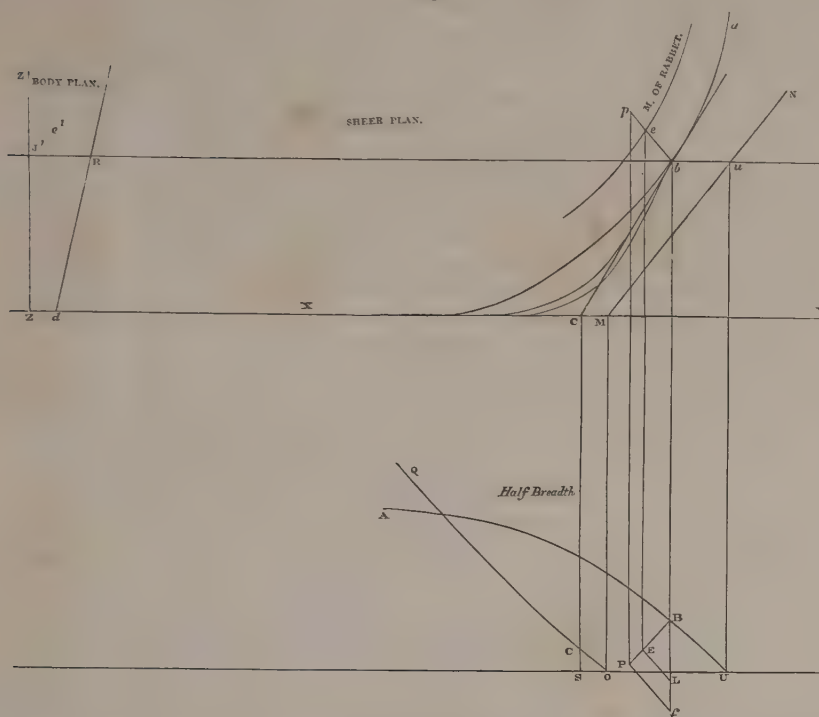
By Rule XV. of the preceding section, draw the projection BG of a perpendicular from the point B on the plane whose traces are LMN : G will be the projection of a point in the middle rabbet. Through G draw FGH parallel to CB , cutting XBM in H ; this point will be the required projection of the point where the middle rabbet cuts the water-section. From H draw HK perpendicular to BM , cutting LM in K ; this point will be the required trace of the middle rabbet on the water-section. Join BK ; BKM will be a horizontal section of the rabbet or groove in the side of the stem. The same process may be applied to the rabbet of the stern-post.

21A. METHOD II.—*Exact Process.*—The most general, as well as the usual case, however, is when the side of the stem is a longitudinal plane inclined to the vertical longitudinal plane, and the curvature of the fore-edge of the rabbet of the stem, in the sheer-plan, is considerable. In such a case it is required to find the middle of the rabbet. This is done by drawing tangent planes to the outer surface of the ship at several points, drawing perpendiculars to those planes through those points equal in length to the thickness of the planking, and finding the projections of the ends of those perpendiculars, which will be points in the middle of the rabbet.

In Fig. 6A let bR represent a level line in the sheer and body plans; b its intersection with the fore edge of the rabbet of the stem; dR and $a\delta$ the projection of the fore edge of rabbet in the body and sheer plans. Through b draw bca tangent to the fore edge of rabbet at b , cutting the axis XX in c ; draw cCS perpendicular to the axis, meeting the axis of the half-breadth plan in S ; make CS equal to the distance dZ in the body-plan; then C is the horizontal trace of the tangent to the fore edge of rabbet at b . Through C draw QCO parallel to the tangent to the water-line in the half-breadth plan at B , meeting the axis in O . QCO is the horizontal trace of the tangent plane to the water-line at B , b . Also draw the tangent to the water-line at B , to cut the axis in U . Through the point U draw Uu perpendicular to the axis, meeting

the water-line, in the sheer-plan, in u ; u is the vertical trace of the tangent to the water-line at b ; also project O to the axis of the sheer-plan in M . A line, MN , drawn through M and u , will be the vertical trace of the tangent plane to the surface of the ship at B , b . Through B and b draw BP and bP perpendicular to the horizontal and vertical traces of the plane, OQ , MN , respectively.

Fig. 6A.



This line is perpendicular to the tangent plane at the point Bb . Take any point, P , p , in this perpendicular, and take for the moment the plane of the level line bR as the horizontal plane of projection. Through P draw Pf perpendicular to BP ; and make Pf equal to the distance of p above the level line bR in the sheer-plan. Join Bf ; Bf is the rabatment on the horizontal plane of a line perpendicular to the tangent plane at Bb . Make BL equal to the thickness of the planking. Draw LE perpendicular to BP ; E is the horizontal projection of a point in the middle of the rabbet. Draw Ee perpendicular to the axis meeting the bP in e ; e is the vertical projection of a point in the middle of the rabbet. The point Ee may next be projected into the body plan to e' . In the same manner other points may be found on the three plans in the middle line of the rabbet, and the traces, on the planes of the level lines, of the curved line passing through those points will be the points at which the several level lines will end. When the several level lines have been drawn in and ended as described, the bearding-line is obtained as follows:—

Draw in the half-breadth plan short lines parallel to the axis at distances from it equal to the half-sidings of the stem (obtained from the body-plan) at the several water-lines; the points in which those short lines cut the water-lines will be points in the bearding-line required. Through those points draw lines perpendicular to the axis to meet the respective water-lines in the sheer-plan. A curve drawn through those points will give the bearding-line required in the sheer-plan.

SECTION II.—PROPERTIES AND USE OF VARIOUS LINES ON THE BUILDING-DRAUGHT.

22. *General Explanations.*—The lines shown on the building-draught may be either imaginary or real. When they are imaginary, they serve, like the corresponding lines on the calculation-draught, simply to show the figure of the ship's skin; when real, they represent also the "moulding edges" of pieces of the framework, and the seams of the pieces of the skin, whether wooden planks or iron plates.

Each piece of the framework of a ship that is in contact with the skin, meets the inner surface of the skin at two edges. One of those edges is shown by a line on the building-draught; it is called the "moulding edge." The plane of the moulding edge of a piece of the frame is called the "moulding plane" of that piece. The other edge is called the "bevelling edge."

The scale of the building-draught, on paper, is usually (in British measures) one quarter of an inch to a foot, or one forty-eighth part of the real dimensions. On the mould-loft floor, it is drawn to the full size. The present Chapter has reference chiefly to the draught on paper; nevertheless all the processes described in it are capable of being performed on the floor; and with respect to many of them it is a matter for the discretion of the

constructor, whether he will perform them on the paper, and copy their results on the floor, or perform them on the floor alone.

Those processes have in general the following objects:—

I. From the lines originally shown on the building-draught, to deduce the figures of other lines more conveniently placed for purposes of laying-off.

II. To construct the "rabatment," or real shape of the moulding edges of pieces of the frame in any required positions.

III. To find the "bevellings," or angles which the sides and edges of each piece of the frame make with each other.

IV. To construct the "expansion" of the skin of the vessel.

V. To construct lines belonging to the inner skin or lining (if any), and to inner pieces of the framework, such as keelsons, deadwood, &c.

All those problems are solved by the proper application of the rules of descriptive geometry, given in Chapter I., Section II., of this Division.

23. *Frames or Bends.*—The cross-sections of the building-draught are called "frames" or "bends," because, when they are real lines, they represent the moulding edges of a series of ribs of timber or iron, which are called by the same names, and which, in the ordinary way of building ships, form the principal part of the framework. They cross the keel perpendicularly, at points which are called "stations."

When the frames are real, the distance from station to station is called "room and space;" according to ordinary practice, it

ranges from about $1\frac{1}{2}$ foot to 4 feet in different vessels; and it is sometimes different in different parts of the same vessel. The determination of what it ought to be is a question of strength, and will be considered in the Third Division.

When the frames are imaginary lines, their distance apart is a matter of convenience; and from 6 to 12 feet is in general close enough.

The "midship bend" or largest frame is sometimes called "dead-flat," and marked in the draught with the symbol, \oplus , as formerly stated. When there is a straight middle body, its equal and similar frames are all called "flats."

According to the longest-established practice, the frames of the fore-body are marked in the draught with letters, A, B, C, &c., and those of the after-body with numbers, 1, 2, 3, &c., commencing in each case with the frame nearest dead-flat (see for example, the sheer-plan of H.M.S. *Victoria and Albert*, Plate $\frac{7}{2}$); but it is now common simply to number them in succession from the bow to the stern (as in the sheer-plan of H.M.S. *Warrior*, Plate $\frac{7}{1}$).

Although every frame must be drawn full-sized on the floor, it is not necessary nor desirable to draw more frames on paper than are sufficient, in the opinion of the constructor, to show the form of the ship with precision. For example, in the plans of the *Warrior* (Plate $\frac{7}{1}$), every third frame is drawn, from No. 34 to No. 97, and every sixth frame only from No. 5 to No. 29, and from No. 103 to No. 127; because the frames are closer together at the bow and stern than amidships.

The frames, being commonly perpendicular to the rabbet of the keel, are oblique to the planes of flotation, if the vessel is not to float on an even keel. In Division First, Article 96A, it has already been explained how such oblique transverse sections are to be used in calculating displacement and stability; it remains to be shown how their form is to be constructed (see Fig. 7).

The adjoining figure shows the construction of the midship frame, \oplus , and two other frames marked 3 and D, and of the projections of three water-lines, on the body-plan of a vessel whose draught of water aft is to be nearly double of her draught forward. On the sheer-plan, the rabbet of the keel is drawn as if it were horizontal, and the water-lines, L.W.L., 2 W.L., 3 W.L., are drawn with their intended slope *relatively to the keel*. The projections of the frames are marked upon the sheer plan by lines drawn as if they were vertical.

On the half-breadth plan are laid down, not the water-lines themselves, but the *projections* of those lines on a plane parallel to the keel; which projections are as easily drawn as the water-lines; for the ordinates are the same in length, and are merely to be placed closer than the real ordinates in the proportion in which the length between the perpendiculars on the rabbet of the keel is less than the length between the same lines measured upon a plane of flotation. (See Article 12 of this Division.) The projections of the gunwale and rail are marked by dotted lines.

The projections of the frames are drawn across the half-breadth plan; and the cross-sections, or figures of those frames on the

body-plan, are constructed by transferring the heights of ordinates above the rabbet of the keel from the sheer-plan, and the lengths of those ordinates, or half-breadths, from the half-breadth plan. (For the sake of distinctness, the frames 3 and D are shown separately in the figure, above their respective stations, as well as forming part of the body-plan.) The projections of the water-lines, and of the gunwale and rail, are marked on the body-plan by dotted lines traversing the ends of the proper ordinates.

The cross-section marked 4 is that of the stern at the after-perpendicular.

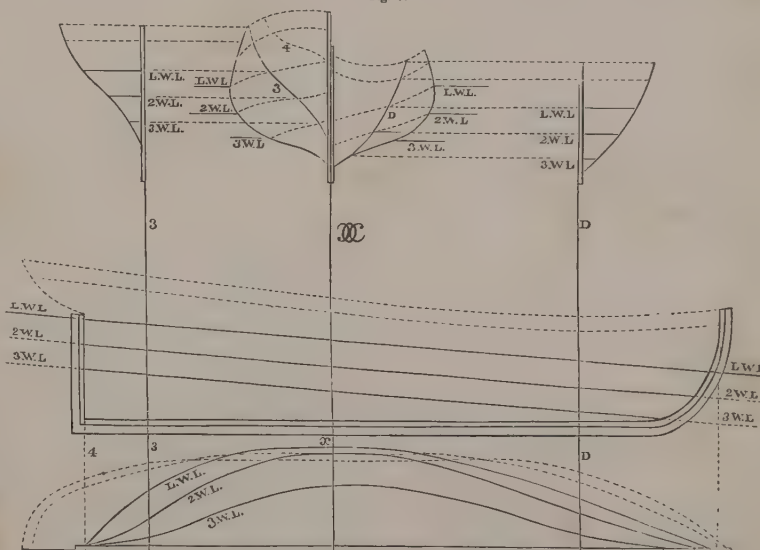
Illustrations of the results of those processes on a large scale are given in the plans of H.M.S. *Warrior* (Plates $\frac{3}{1}$, $\frac{3}{2}$), and of H.M.S. *Victoria and Albert* (Plate $\frac{9}{1}$).

24. *Level Lines* is the name given to sections of the inner surface of the ship's skin by planes which are really level athwartships, and parallel to the rabbet of the keel longitudinally. If the ship is to float on an even keel, they are identical with water-lines. In other cases, they are easily constructed on the half-breadth plan, when a sufficient number of frames are given on the body-plan, by drawing horizontal straight lines on the body-plan to represent the traces of planes parallel to the keel, and transferring half-breadths from the frames on the body-plan to their proper stations on the half-breadth plan.

For many purposes it is sufficient, instead of drawing a level line entire, to draw tangents to level lines at a series of given points. That problem is solved by means of the following—

RULE.—Given the slope, on the sheer-plan, of the water-lines

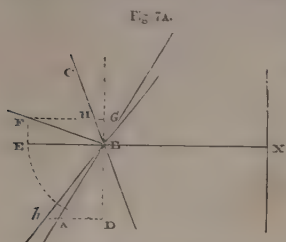
Fig. 7.



relatively to the level lines; the projection, on the half-breadth plan, of a tangent to the water-line traversing a given point; and on the body-plan, a tangent to the frame traversing the same point: to draw on the half-breadth plan a tangent to the water-line which traverses the same point.

Let Fig. 7A represent part of the body-plan; B, the given point, at the end of the ordinate XB, and BC a tangent to the frame that traverses B. Transfer from the half-breadth plan the angle XBA, which the projection of a tangent to the water-line at B makes with the ordinate XB. In BA take any convenient point, A, and draw AD parallel and BD perpendicular to XB, meeting

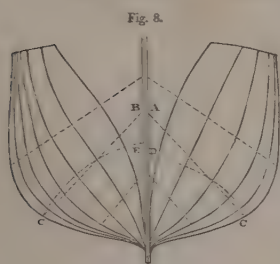
each other in D. In XB, produced if necessary, lay off BE = BD. Transfer from the sheer-plan the angle, EBF, which the water-lines make with the level lines. Perpendicular to BE, draw EF cutting BF in F; and through F draw FG parallel to EBX, cutting BC in H and DB produced in G. On DA, produced if necessary, lay off Ah = GH, and join Bh; XBh will be the angle between the ordinate XB and the tangent to a level line at B.



To prevent mistakes as to the direction in which Ah is to be laid off, it is to be borne in mind, that at all points where the side *flares out*, in a vessel which floats *by the stern*, XBh is *greater* than XBA for the lines of the *after-body*, and *less* than XBA for the lines of the *fore-body*. At points where the side *tumbles home*, those relations are reversed; and they are also reversed if the vessel floats by the head. At all points where the side is vertical XBh is equal to XBA.

25. *Diagonals or Riband-lines* are sections of the inner surface of the ship's skin, made by planes which are oblique in a thwartship direction, and longitudinally parallel to the rabbet of the keel. They are used in the same manner with water-lines and buttock-lines (which they resemble in shape), for testing the fairness of the ship's form, and for constructing the figures of frames. The positions of the diagonal planes are laid down on the body-plan so as to cross as many frames as possible nearly at right angles, and so as to divide the frames into parts of nearly equal lengths; and in wooden vessels, their positions are chosen with a view also to their passing through convenient points for the futtock heads, or junctions of the pieces of timber of which the frames are built; a subject which will be further considered in the Third and Fourth Divisions.

The body-plan, Fig. 8, shows, by straight dotted lines, the traces



of three diagonal planes on the fore-body and after-body respectively; AC and CB being respectively the forward and the after part of the same diagonal.

The "rabatment," or real figure, of a diagonal line, is constructed on the half-breadth plan, simply by transferring oblique ordinates from the traces

of the diagonal plane on the body-plan, to the proper stations of the frames on the half-breadth plan.

It is usual also to give the *projections* of the diagonals upon the sheer-plan and half-breadth plan respectively. The latter projections are called "horizontal ribands."

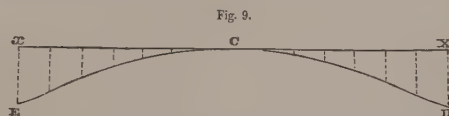
26. The term *Normal Lines* may be applied to curves which cross all the frames at right angles. When the frames are really vertical, these are the *geodetic lines* already mentioned in Article 160 of the First Division.*

An originally straight batten, being applied to the real frame of a ship so as to cross the midship frame at right angles, and then

bent so as to touch the other frames, assumes of itself the form of a normal line; so also does an originally straight spring, when applied in the same manner to the model of a ship.

To construct such a curve on paper, its projection on the body-plan must first be drawn. That projection consists of a pair of curves on the fore-body and after-body respectively (as CD and CE in Fig. 8) which start from the same point, C, in the midship frame, and cut all the frames at right angles. A practised draughtsman may sketch them by the eye with considerable accuracy; but if great precision is required, the process is as follows:—At C, draw a tangent to the midship frame, and a perpendicular to that tangent, or *normal*; this normal will be a tangent to the required projection. Take a point in that normal midway between the midship frame and the next frame, and about that point, with a radius a little greater than the distance to the next frame, draw a small circular arc cutting the last mentioned frame in two points; bisect the interval between those points, for a second point in the curve. Starting from that second point as from the point C, drawing a tangent and a normal to the frame as before, and repeating the same operations, a third point is found, in the second frame from the midship frame; and thus by repeating the process, are found a series of points at the intersections of the required curve with every frame in the body-plan, together with tangents at those points; from which data the required projection is easily completed.

To construct the *Development of a Normal Line* (see Article 15 of this Division), draw an axis xCX (Fig. 9), to represent a tan-



gent to the normal line at the point C where it crosses the midship-section, which tangent is parallel to the rabbet of the keel. Lay off CX and Cx to represent the lengths of the fore-body and after-body respectively, and mark upon the tangent XCx the stations of the several frames. Then, by means of a suitable instrument,† measure ordinates *round the curve* to the frames on the body-plan, from C towards D for the fore-body, and from C towards E for the after-body, and set up the ordinates so measured at the stations of the frames to which they belong on the tangent xCX. The curve ECD drawn through the ends of those ordinates will be the required development; its curvature, and its inclination to its axis, at each point, will be the same as those of the normal line at the corresponding point; and its length between any two of its points will be the same with the length between the corresponding points on the normal line.

A longitudinal frame of an iron ship takes nearly the path of a normal line, and the development of such a frame may be obtained in the manner above described.

27. *Sheer-lines*, comprehending the top-side and gunwale, the port-sills in ships of war, the lines where the under surfaces of the planking of the decks meet the skin of the ship, and various seams in the skin, are in general curves of double curvature, whose projections on the sheer-plan and half-breadth plan form part of

* These lines are called "dividing lines," by Lord Robert Montagu, who has shown the importance of their fairness.

† A simple instrument for measuring the lengths of curves, consisting of a small wheel with a milled or spiked edge, turning upon a screwed spindle, gives very accurate results when well made and carefully used. On the mould loft floor a batten may be bent round the curve of the Normal Line, and the intersections of that curve with the square sections may be marked on the batten.

the original design of the vessel; so that their projections on the body-plan are constructed simply by transferring heights from the sheer-plan, and half-breadths from the half-breadth plan.

The *development* of any such curve may be constructed, if required, by the process already described in the preceding Article, provided the curve, at its greatest breadth, is parallel to the keel; and such is generally the case in practice.

Should the curve to be developed, at its greatest breadth, be inclined to the keel at any considerable angle, the process requires the following modification:—Increase the length of the tangent αCX , Fig. 9 (which serves as the axis of the development), and of each interval on the tangent between the stations of the ordinates, in the ratio of the secant of that angle to radius. The ordinates of the development are to be measured, as in Article 26, round the projections of the curve on the body-plan: the development will not be absolutely exact, but it will be accurate enough for practical purposes, such as measuring the length of the curve, or of any part of it.

28. A *Cant Frame* is a frame whose moulding edge is situated in a plane perpendicular to the level-line planes, but oblique to the plane of the sheer-plan. Such frames are used in the foremost and aftermost parts of vessels of a somewhat full form, in order to be less oblique to the skin than *square frames*, or frames perpendicular to the rabbet of the keel; and in some recent systems of shipbuilding, all the frames are cant frames.

The traces of the plane of a cant frame are, on the sheer-plan, a vertical line, Zc (Fig. 10), and on the half-breadth plan, an oblique straight line, AB .

The rabatment of a cant frame may be drawn on the sheer-plan, by the aid either of level-lines, or of water-lines not coinciding with level-lines, intersecting the trace, AB , on the half-breadth plan. If the lines on the half-breadth plan are level-lines, all that is necessary is, to lay off the heights of those lines upon Zc in the sheer-plan, and draw horizontal ordinates, whose lengths are taken by measurement along AB on the half-breadth plan. If the lines on the half-breadth plan are water-lines not parallel to the keel, proceed as follows:—Let ab be the trace, on the sheer-plan, of the water-line which traverses B on the half-breadth plan. Draw Bb parallel to aA , cutting ab in b ; then b will be the projection, on the sheer-plan, of the point whose projection on the half-breadth plan is B ; and ab and AB will be the two projections of one oblique ordinate of the cant frame, situated in the plane of flotation, ab . In the same manner the projections of other points may be found, so as to give, if required, the vertical projection, Zb , of the cant frame. Through b draw cd parallel to AX (the axis of projection), and equal to AB ; d will be the rabatment of the point in the cant frame whose projections are B and b ; and ad will be the true length of the oblique ordinate whose projections are AB and ab . A series of points found in the same manner enable the true figure of the moulding edge, Zd , to be constructed.

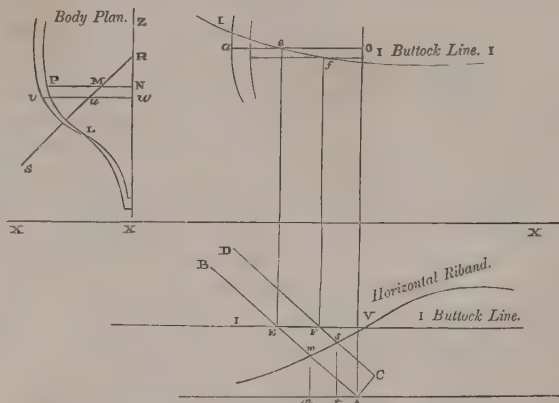
In the same manner may be constructed the moulding edge of any piece whose moulding plane is similarly placed to that of a cant frame.

In wooden ships it is necessary to draw the diagonal lines and their projections on the mould-loft floor; and after the body-plan has been faired, these are the only lines remaining on the floor.

The cant timbers are therefore usually laid off by means of the projections of the diagonals, or horizontal ribands, and in the following manner:—

Let mV , Fig. 10A, be a horizontal riband, m the point in which it is cut by the horizontal trace AB of the cant; and let RS be the diagonal in the body-plan. Take the perpendicular distance mn of the point m from the middle line of the half-breadth plan, and find the point M in the diagonal RS in the body-plan where MN , perpendicular to the middle line, is equal to mn . Produce NM outwards and make NP equal to the distance Am along AB of the point m from the middle line of the half-breadth plan; the point P thus found will be a point in the rabatted cant section.

Fig. 10A.



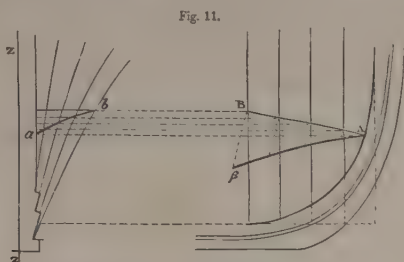
In the same manner a point in the cant frame is obtained at every diagonal. Points in the cant section may also be obtained by means of buttock or bow lines in the following manner:—Let I, I , Fig. 10A, represent a buttock-line in the half-breadth and sheer plans. Through the points A and E where the cant cuts the middle line and buttock-line respectively, draw AO and Ee perpendicular to the axis, Ee cutting the buttock-line in the sheer in e . Through e draw OeQ parallel to the axis, cutting AO in O , and make OQ equal to AE of the half-breadth plan. Then Q is a point in the cant section. In the same manner points in the cant section may be obtained at all the buttock or bow lines.

29. *Breast-hooks and Transoms*.—Breast-hooks at the bow, and transoms at the stern, are frames whose moulding planes are horizontal or slightly sloping (those in way of the decks have their upper or moulding surfaces cylindrical, with a round-up equal to that of the under side of the decks), and which spring from the stem and sternpost in the same manner that the square frames spring from the keel.

If the moulding plane of the piece is horizontal, its moulding edge is simply a water-line; and if parallel to the keel, a level line.

When the moulding plane is inclined, the construction of the transverse projection and the rabatment of the piece are illustrated in Fig. 11, where the right-hand division of the figure represents part of the sheer-plan, at the bow of the vessel, and the left-hand division, part of the fore-body plan. AB is the trace on the sheer-plan, of the moulding-plane of a breast-hook, intersecting a series of frames. Draw Aa , Bb , and the lines between them, from the points of intersection parallel to the axis of projection; the points where those parallel lines cut the successive frames in the body-plan will be points in the transverse projection, ab , of the breast-hook.

Draw ordinates on the sheer-plan perpendicular to AB , at the points where it cuts the frames, and transfer the lengths of those ordinates from the body-plan; the curve, $A\beta$, through their ends



will be the rabatment of the moulding edge required. Transoms of a sharp figure are constructed like breast-hooks, by the aid of the frames shown on the body-plan. Those of a bluff figure are sometimes more accurately constructed by the aid of buttock-lines; but the principle of the process is the same in either case. When the upper surface of the transom coincides with the under surface of the deck, the intersections of the several buttock or bow lines, and the cylindrical surface of the upper side of the transom, are obtained in the sheer-plan; and the points in which these lines cut the respective buttock or bow lines will be points common to the cylindrical surface of the transom and the surface of the ship. The cylindrical surface is then developed in the manner described for the normal lines.

30. The "*Scanlings*," or transverse dimensions of pieces, are called "*moulding*" and "*siding*;" *moulding* being the depth or dimension which lies in the moulding plane; and *siding*, the thickness in a direction perpendicular to the moulding plane. For example, if a piece of timber in a frame is said to be "moulded 12 inches, sided 14 inches," that means, that the piece in question measures 12 inches deep on its moulding plane, and 14 inches thick in a direction perpendicular to that plane.

In an angle-iron or L-shaped iron rib, the moulding plane is formed by the outer face of one of the two flanges which meet at the angle. The outer edge of the angle is the moulding edge; and the siding may be held as represented by a perpendicular let fall on the moulding plane from the farther edge of the other flange, or bevelling edge.

The moulding of the keel, stem, and sternpost, is measured in the plane of the sheer-plan, and the siding, directly athwartships. The moulding of a deck-beam is its depth, and the siding its breadth.

The term "*Bevelling*" denotes the angle which two surfaces make with each other at the line where they intersect. In speaking of the pieces of the frame of a ship, it is applied to the angle made by the surface at which the piece touches the skin, with the moulding plane of the piece.

When the moulding plane is parallel to one of the planes of projection (as in square frames), the bevelling is found by the use of Rule VIII., Article 10. When the moulding plane is not parallel to the plane of projection, the bevelling is found by the use of Rule XII.

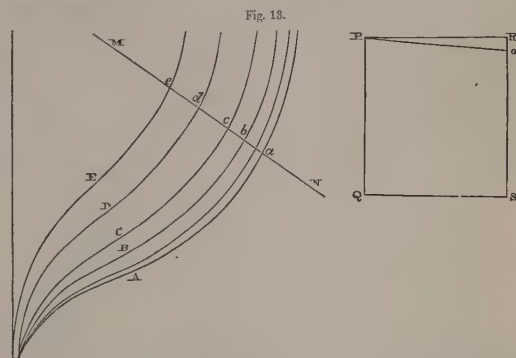
For example, let Fig. 12 represent part of the body-plan, XB the ordinate at the point B , and BC the tangent to the moulding edge of a square frame at B . Transfer, from the half-breadth plan, the angle, XBA , which the tangent to the level line at B makes with the ordinate; then conceive XB to be the axis of projection, so that AB and BC may represent the traces of a tangent plane at B ; construct by Rule VIII. the angle which that

plane makes with the vertical plane of projection; and that angle will be the required bevelling.

Another way of finding the bevellings of square frames is by constructing the development of normal lines, which cross their moulding edges at right angles, as described in Article 26 of this Division; for the angle made by such a line with its ordinate at a given point is the exact bevelling of the square frame at that point.

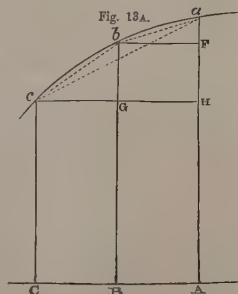
The common method of obtaining the bevellings of the frames in the square body is as follows:—For example, if it be required to find the bevellings of the square frames Aa , Bb , Cc , Dd , &c., at the points a , b , c , &c., in the body-plan, Fig. 13—

From the point a take the shortest distance to the square frame next to it. In a convenient place draw two parallel lines, PR and RS , the distance PR between them being equal to the space between the square frames. Draw PR perpendicular to PQ or RS ,



meeting RS in R ; and from the point R set off the distance Ro equal to the shortest distance from the point a to the square section next outside it; join Po , and the angle SoP will be the bevelling of the square frame, Aa , at the point a . In the same manner by setting off from R the shortest distances from the points b , c , d , &c., to the square sections respectively next outside of them, joining the points thus found with P , the bevellings of the several square frames at b , c , &c., will be found.

If the curvature of the normal line at the bevelling point be very great, the following method is more accurate:—Through the bevelling point draw a line which is the shortest distance between the two square sections adjacent to that in which the bevelling point has been taken. From the point R , in the line RS , set off half the shortest distance thus found between the two square sections adjacent to that in which the bevelling point is taken; join this point and P , and the angle which this line makes with PQ will be the required bevelling.*



* Let Fig. 13A represent the section of the ship by a plane passing through the point b normal to the square section at that point. Aa , Bb , and Cc , represent square frames. Through c and b draw cGH and bPF parallel to AC ; join bc , ba , and ac . Then the bevelling obtained by the first method at the point b , is the angle Aab ; and by the second method the angle, AaC . And since the chord ac is nearly parallel to the tangent at b , the latter method gives a closer approximation to the bevelling of the square frame at b .

Should the actual frame at B be a cant frame, draw BC and BA as before, BC being now the tangent to an *imaginary* square frame at B. Draw the vertical line, BD, and transfer from the half-breadth plan to the body plan the angle, XBE, which the cant frame at B makes with the ordinate, XB; then, as before, conceive XB to be the axis of projection; ABC will be the traces of the tangent plane at B, and EBD the traces of the moulding plane of the cant frame. Construct, by Rule XII., the angle between those planes; that angle will be the required bevelling.

The most convenient method, however, of obtaining the bevelling of a cant timber is to project the bevelling edge on the plane of the moulding edge, and rabat the plane of the moulding edge about its vertical trace.

For that purpose draw AC (Fig. 10, page 119) perpendicular to AB in the half-breadth plan, make AC equal to the siding of the cant timber, and draw CD parallel to AB. CD is the bevelling edge of the cant timber.

To obtain points in the bevelling edge by means of level-lines, it is only necessary to take the distances of the intersections of the bevelling edge, CD, and the level-lines in the half-breadth plan from the point C (Fig. 10), and set them off from the axis AZc on the corresponding level-lines in the sheer-plan. A curve drawn through these points will give the projection of the bevelling edge on the plane of the moulding edge; the plane of the moulding edge having been rabatted on the sheer-plan about its vertical trace AZc.

If the lines on the half-breadth plan are water-lines not parallel to the keel, through the intersections of the bevelling edge with the water-lines in the half-breadth plan draw lines perpendicular to the axis, AX (Fig. 10, page 119), meeting the respective water-lines in the sheer-plan. Through the points thus obtained draw lines parallel to AX: on these level-lines set off distances equal to those of distances of the intersections in the half-breadth plan of the respective water-lines and the bevelling edge from the point C. Through the points thus obtained the bevelling edge will pass.

The lines in which the plane of the bevelling edge intersects the water-lines might also be obtained as were those for the moulding edge; but those lines are not required.

To obtain points in the bevelling edge by means of the horizontal ribands: in Fig. 10A, page 119, draw in the bevelling edge as already described.

Take the distance *st* from the middle line (Fig. 10A) of the point *s* in the half-breadth plan where the bevelling edge cuts the horizontal riband; and as before done for the moulding

edge, find the point *u* in the diagonal of the body-plan, whose horizontal distance, *uw*, from the axis is equal to *st*.

Produce *wu* if necessary, and upon it set off *wv* equal to the distance *Cs* in the half-breadth plan; the point *v* is a point in the bevelling edge when projected on the plane of the moulding edge, and when the plane of the moulding edge is rabatted about its trace on a transverse vertical plane. The moulding and bevelling edges being thus obtained, the bevellings of the cant are found, as already

described for the square body, as follows:—In a convenient situation draw two parallel lines, AB and CD (Fig. 13a), their distance apart being equal to the siding of the

cant timber, and draw AC perpendicular to AB. At the points in the moulding edge where it is required to obtain bevellings, take the shortest distance to the bevelling edge, and according as the bevelling edge is inside or outside of the moulding edge set off on the line CD the shortest distances, as *Cx*, *Cy*, above or below the point C; join *Ax*, *Ay*, and the angles, *xAB*, *yAB*, will be the bevellings required. It is clear that at the point L (Fig. 10A, page 119, where the moulding and bevelling edges intersect, the bevelling is a right angle.

The bevelling of a transom, the upper surface of which coincides with the under side of the deck, is obtained in the sheer-plan by drawing through the intersections of the upper edge of the transom, and the several buttock or bow lines, lines parallel to the sheer of the deck; the angles between these lines and the buttock-lines will be bevellings of the transom at the respective buttock or bow lines. In setting off the bevelling on the timber, care must be taken that the stock of the bevel is fore and aft, and its tongue perfectly vertical.

31. *Expansion of the Skin*.—The nature of an expansion of a surface (or rather of a certain network of lines on that surface) has been explained in Article 15 of this Division. The process of expansion is applied to the skin of an intended ship, in order to facilitate the laying-off of the dimensions and positions of the pieces of which that skin is to be made, whether timber planks or iron plates.

Those pieces are fastened upon the frames in long bands called *strakes*, meeting each other at long joints called "*seams*," and divided into lengths by short joints called "*butts*," at right angles, or nearly at right angles, to the seams. The skin of each side of the vessel is bounded in front by the rabbet of the stem, below by the rabbet of the keel, above by pieces called the "*gunwale*" or the "*plank-sheer*," according to their position, and abaft, as high as the rudder-post, by the rabbet of the stern-post. In some vessels the upper part of the skin is bounded abaft by a rabbet in a straight or slightly curved and nearly horizontal piece, called the "*tuck-rail*," and in a piece at each side, slightly raking aft, called the "*side counter-timber*," which pieces form the lower and lateral boundaries of the ornamental work of the stern; in other vessels, the skin goes continuously round the upper part of the stern.

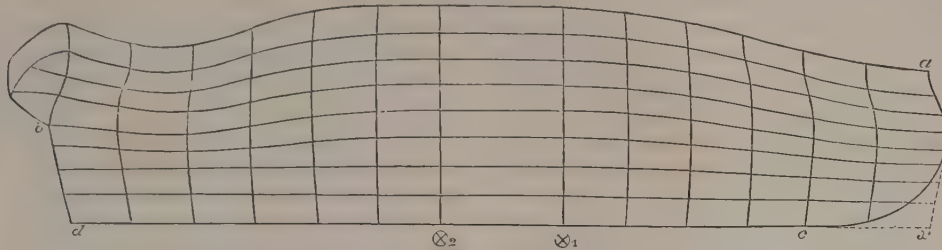
The details of the expansion of the skin will be described further on; the object of the present Article is to show how to construct a skeleton diagram or network, giving a sufficient number of lines to guide the draughtsman afterwards in laying off the details of planking or plating.

According to the ordinary construction of ships, the seams of the skin are nearly longitudinal, and most of them run continuously from stem to stern, the lowest being where the lowest strake, called the "*garboard strake*," fits into the rabbet of the keel; and the butts are parallel, or nearly parallel, to the adjoining frames. Fig. 14 represents an expansion-diagram suitable to that style of building. The nearly upright lines represent frames; the longitudinal curves, seam-lines; *ac* is the mid-rabbet line of the stem, *cd* that of the keel, and *db* that of the stern-post; and the curved boundary rising above *b* is the centre line of the skin of an elliptical stern. The body-plan of the vessel to which the expansion diagram belongs is the same with that of the ship represented in Division First, Chapter VI., Fig. 1; but the length of that ship is supposed to be first shortened to two-thirds, and then lengthened by cutting it in two amidships, and inserting a straight middle body between the marks \oplus_1 and \oplus_2 . The midship frames are represented by vertical straight lines.

For two reasons, it is important that the seam-lines of the bottom should as nearly as possible coincide with the *normal lines* described in Article 26 of this Division, as crossing all the square frames at right angles; first, because when such is the case, the

strakes of plank or of plate, when bent to fit the bottom, are curved *flatwise* only, and have no curvature *edgewise* (which curvature, when concave upwards, is called *sny*, and when concave downwards, *hang*);^c and secondly, because the particles of water

Fig. 14.



in general tend to follow nearly the course of normal lines; so that the overlapping edges of the strakes of clinker-built vessels (such as small wooden boats, and iron vessels of all sizes) cause on the whole less resistance in that position than in any other.

The seams of the upper part of the side must follow the sheer of the top-side, gunwale, portsills, and other sheer-lines.

To make the expansion-diagram, then, the constructor commences by drawing on the body-plan the projections of a series of sheer-lines, to guide the seams of the side; and of normal lines, or lines approaching as nearly as practicable to them, to guide the seams of the bottom. He next, by the methods described in Articles 26 and 27, constructs the *development* of those seam-lines. Then by measuring distances on the body-plan, from the mid-rabbit of the keel round the frames to their points of intersection with the seam-lines, and on the development-drawings round the seam-lines, from the midship-frame to the same points, he is enabled to construct such a network as that shown in Fig. 14.

When there is a middle-body, its expansion is a true development; and the normal lines upon it are simply parallel horizontal straight lines.

32. The *cutting-down line* is a line on the sheer-plan, which touches the lowest part of the inner surface of each of the frames. It is nearly similar in form to the stepping-line, above which it is situated at a height equal to the vertical depth of each frame at its lowest part. That vertical depth may be measured on the body-plan, when the moulding of the frames is given, by drawing in the first place the *inner edge of the moulding side* of each frame, and then measuring its height above the outer or moulding edge in the vertical plane of the side of the keel.

But if the vessel is to have a continuous *inner skin* or lining, whether of timber or iron, the dimension to be used in constructing the cutting-down line may not be the moulding of the frames, but

the *perpendicular clear distance* between the outer and inner skins; and in that case the cutting-down line is to be laid off from the stepping line, precisely as the stepping line is laid off from the rabbets, by the method described in Article 18 of this Division.

In wooden ships, the forward and after parts of the cutting-down line sometimes mark the upper edge of a rabbet in the dead-wood, whose lower edge is the stepping line, and from which the cant frames spring.

The midship part of the cutting-down line in most ships runs along the lower surface of the *keelson*, which is parallel to and directly above the keel, and bears the same relation to the inner skin (if any) that the keel does to the outer skin.

33. *Inner Skin and Deck-Lines.*—*Inboard Works.*—Lines either on the outer or the inner surface of an inner skin, or on any surface parallel to and at a given distance within the outer skin, may be constructed, when required, by the processes already described in the preceding Section, which serve to deduce the figures of lines in the building-draught from those of lines in the calculation-draught.

The sheer-lines of the decks, marking where the under surface of the planking of each deck meets the inner surface of the frame, belong to this class; but they are easily drawn on the sheer-plan without any special process, except such as may be required in making them fair curves from head to stern.

The *Round-up* of the upper side of the deck-beams, being a convexity given to them for the purpose of causing water to run off the decks, may be shown on the body-plan, if required. It is usually from 0.01 to 0.02 of the extreme breadth of the vessel, being proportionally largest in the smallest vessels.

The *Inboard Works* of various kinds are shown on longitudinal and cross-sections, and on deck-plans. Several examples of such works are given in the Plates.

CHAPTER III.

ON LAYING-OFF AND TAKING-OFF.

34. *Nature and Object of Laying off.*—The operations of laying-off consist partly in making full-sized drawings of the frame and skin of a ship upon the floor of the mould-loft, and partly in the construction of *moulds* and *beveling-boards* for various pieces of the frame; *moulds* being full-sized patterns, of the same figures and

dimensions with the moulding sides of the pieces which they represent, and *beveling-boards*, flat pieces of wood on which the

^c A comparison of the curvature of one of the projections of any given line on a ship's skin, with the curvature of the corresponding projection of the nearest normal line, shows at once whether the given line "hangs" or "snyes."

bevellings of the several pieces are marked. The object of those operations is to provide full-sized representations of the several parts of the ship, by the aid of which, in the first place, pieces of timber and iron suitable for making those parts may be provided; and in the second place, those pieces may be made to their true dimensions and figure; so that when put together, they shall form a ship exactly agreeing with the design of the naval architect.

35. *Full-sized Drawings* are made on the carefully levelled and planed floor of a large room called the "Mould Loft." Lines for temporary purposes are usually drawn with chalk; lines which are to be permanently used are "rased" or "scribed" on the floor with a pointed tool called a "scriber." The drawings on the floor consist of precisely the same plans, and show the same kinds of lines, with the building-draught on paper, described in the preceding Chapter; and the operations performed in drawing them are in almost every respect, except the scale, the same with those performed in drawing the building-draught, and described in the two preceding Chapters of this Division. In the present Chapter, therefore, the matters which mainly require attention are those in which the full-sized drawings differ from the building-draught. These are chiefly the following:—

I. *INTERMEDIATE FRAMES*.—On paper, it is not necessary to draw more frames than are sufficient to show accurately the figure of the vessel; whereas in the full-sized drawings, the moulding edge of every frame must be shown. (This has already been mentioned in Article 22.) Consequently, after the frames shown in the building-draught have been copied in the full-sized drawing, the stations of the intermediate frames must be marked in the sheer-plan and half-breadth plan, and their figures drawn in the body-plan by the aid of a sufficient number of water-lines, level-lines, buttock-lines, or riband-lines. In the ordinary practice of shipbuilders, riband-lines are the most frequently employed for this purpose (see Article 25 of this Division). The spot where a riband-line cuts a frame is called a "*sirmark*."

II. *FAIRING THE BODY*.—The greater scale of the full-sized drawing makes any want of fairness in the ship's lines more conspicuous in it than in the building-draught. After, therefore, a set of lines requiring fairness, whether water-lines, level-lines, buttock-lines, or riband-lines, have been copied in chalk from the building-draught on to the mould-loft floor, they are to be "faired" before being rased in; that is to say, any small irregularities of figure which were invisible on the paper are to be smoothed away by the eye, or by the aid of an elastic batten. The ordinates of the lines as thus faired are to be used in constructing the full-sized body-plan. A batten should never be left pinned round a curve for a longer time than is absolutely necessary, lest its elasticity should be impaired.

III. *ADDITIONAL RIBAND-LINES*.—The full-sized drawing must show not merely the figures of the parts of the frame of the ship, but the arrangement of the pieces of material of which those parts are to be built up, especially when that material is timber. For example, each of the ribs or frames of a wooden ship consists of several lengths of timber, called floor-timbers, futtocks, and top-timbers, connected by means which will be explained in the Fourth Division. The joints where those lengths are connected together, are so arranged as to lie in a series of diagonal lines; and there are thus as many diagonal lines as there are pieces in a

rib, less one. It may not be necessary to draw all those diagonal lines on paper; but on the floor they must all be drawn.

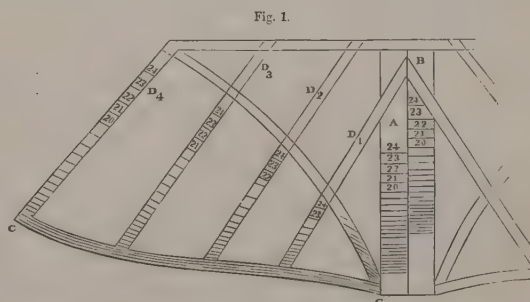
IV. *VARIOUS ADDITIONAL LINES*.—It is seldom necessary or desirable to show on the building-draught all the lines representing the moulding edges of such pieces of the frame as cant frames, counter timbers, breast-hooks, transoms, &c., mentioned in Articles 28 and 29 of the preceding Chapter; but all of them must of course be laid off on the full-sized drawing.

V. *ARRANGEMENT OF PLANS*.—The mould-loft floor is seldom large enough to show the plans of a vessel arranged in the same manner as on paper; and hence their arrangement has to be modified in order that they may be contained within the available space. For example, the half-breadth plan on the floor, instead of being placed below the sheer-plan, has its base-line so placed as to coincide with the base-line of the sheer-plan; and the lines of those two plans are thus mingled together. The two halves of the body-plan, too, are sometimes drawn within one-half the moulded breadth, by being so arranged that the line representing the upright axis in each half of the body shall coincide with the vertical line touching the other half at its extreme breadth. The sheer-plan and half-breadth plan are divided into as many divisions as the length of the ship compared with that of the floor may render necessary; and each such division has a few stations at each end common to it and to the adjoining division, to facilitate the fairing of the lines.

VI. Some differences in the manual operations of drawing on the floor and on paper arise from the difference of scale. For example, when straight lines on the floor are too long to be drawn with a straight-edged ruler, they are marked by means of a chalked cord; and when circles are too large to be drawn with the compasses, methods are employed which are described in Article 6A, and in an addendum at the end of this Division.

If the ordinates of a ship are found by *calculation*, the full-sized drawings may be laid off at once from a table of calculated ordinates, without the necessity of copying from a drawing on paper.

36. *Moulds*, or full-sized patterns of the pieces of a ship's frame, are often, for the sake of lightness, composed of a skeleton framework of battens, having just strength and stiffness enough to preserve its correct figure; and in the case of frames, one mould

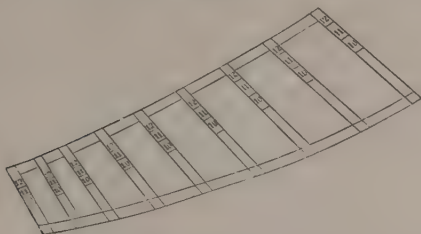


is so contrived as to serve for several frames. For example, Fig. 1 represents a *floor-mould*, which is a portable copy of the lower part of the full-sized body-plan. It consists of two similar halves, one only of which is completely shown, hinged together at the vertical joint between the pieces, A and B, which joint represents the longitudinal midship plane of the ship. The breadth of each of the pieces, A and B, represents the half-siding of the broadest

part of the keel. The transverse lines marked upon the piece, A, and numbered, show the heights at which a series of frames, numbered consecutively up to 24, spring from the rising-line or stepping-line (Article 18); the corresponding transverse lines on the piece, B, show the heights at which the inner surfaces of the same frames meet the cutting-down line (Article 32). The lower edge of the batten, C C, represents the moulding edge of the floor part of the midship frame. The upper edges of the diagonal battens, D₁, D₂, D₃, D₄, represent riband-lines on the body-plan; and upon these battens are drawn and numbered lines marking where they are crossed by the series of frames already mentioned.

Fig. 2 exemplifies a *futtock-mould*, being a portable copy of a higher portion of the full-sized body-plan. The upper edges of the diagonal battens represent riband-lines, as before; the outer

Fig. 2.



edges of the two curved battens represent the moulding edges of portions of two frames; and the figures of the frames intermediate between those two are shown by the numbered lines marked across the diagonal battens.

When a part of the ship's frame is built up breadthwise as well as lengthwise of several pieces of timber (as is the case with the deadwood of the stem and stern), the mould is a flat board whose outline is the same with that of the part represented by it, and on which are drawn lines showing the "shift"—that is, the arrangement of the pieces of which that part is to be built.

37. *Bevelling-Boards*.—The "bevellings" of a ship's frames, at a sufficient number of points, having been determined as described in Article 30 of this Division, are marked upon a series of "bevelling-boards," such as that represented in Fig. 3. The distance between the parallel lines, A B and C D, is equal to the half-siding of the frames to which the board belongs; and the numbers of those frames are marked opposite to a series of lines, which make angles with A B equal respectively to the bevellings of those frames at some particular riband-line. Each bevelling-board is marked with a number or letter showing the riband-line to which it belongs.

38. *Expansion of Skin*.—To draw a full-sized expansion of the ship's skin, a skeleton diagram is, in the first place, to be prepared by the method described in Article 31; and parallel to the longitudinal and transverse lines of that diagram respectively, are to be drawn the longitudinal joints or "seams," and transverse joints or "butts," of the planking or plating, as the case may be; the details of the arrangement or "shift" of that planking or plating being a matter which will be further discussed in the Fourth Division.

An example of such an expansion, for part of an iron ship (the *Persia*), is given in Plate A.

To measure the length of a curved line on the mould-loft floor, a

batten is pinned along the line, and has marks made upon it at the two ends of the part to be measured. The batten is then set free, and the distance between the marks is measured when straight.

39. *Laying-off from a Model*.—The general construction of a model, and its use in designing a ship, have already been described in Article 202 of the First Division.

A model such as has been there described, representing one-half of the ship, bounded by the longitudinal midship plane, and built up of horizontal layers of wood, may be and often is used in laying off a ship, without the aid of a draught upon paper.

The curved surface of a model to be used in laying off usually represents the *inner* surface of the ship's skin, and the dimensions of the model represent the *moulded* dimensions of the ship.

The vertical side of the model, as formerly stated, represents the sheer-plan; and upon it are to be drawn a series of vertical lines, representing the stations of a number of frames sufficient to determine correctly the figure of the vessel, agreeably to what has been stated in Article 22 of this Division. The sheer-plan, as thus completed, is now to be drawn of the full size on the mould-loft floor. In the remainder of the process, two methods are followed, according as the layers of the model are pinned or glued together.

METHOD I.—When the layers are *pinned* together, so that they can be taken asunder, the first operation is to detach them from each other, and to draw, upon their horizontal surfaces, ordinates at the stations of the frames already marked. These ordinates, being measured by the proper scale, give the *half-breadths* at the points where the several frames intersect the several water-lines and sheer-lines; and those half-breadths are to be written in a suitably arranged Table.

From the table of half-breadths, the water-lines and sheer-lines are now to be constructed of the full size, upon the half-breadth plan, and *faired* by the use of battens. The process of fairing may slightly increase or diminish some of the half-breadths; and the half-breadths as thus faired are to be entered in an amended Table.

The full-sized body-plan is then constructed with the half-breadths as faired; and the remainder of the process of laying-off presents nothing peculiar.

The Tables of half-breadths may be used in the computation of moulded displacement, and in other calculations.

METHOD II.—If the layers of the model are *glued* together (which is more favourable to fairness of the model than pinning them), the following method is employed.

The model is laid with its flat side on a level board. Upon that board a frame is placed, which bestrides the model, and is capable of being slid along the model longitudinally into any required position. Part of that frame consists of a guide, in which a pencil moves so as to be always in a plane perpendicular to the keel of the model. By shifting the frame into a series of different positions, and using the pencil, a series of cross-sections are drawn upon the curved surface of the model.

Those sections are then copied, so as to construct the body-plan either on paper or on the mould-loft floor, by measuring with callipers either their half-breadths at the several water-lines, or the distances of the points where they cut the several water-lines from the keel. The body is then faired as in Method I.

Normal Lines are easily drawn, if required, on the surface of a model by means of a spring, or of a whalebone batten, agreeably to the principles stated in Article 26.

40. By *Taking-off* is meant, the operation of performing such measurements upon an actual ship as shall enable her plans to be drawn. The process described in the preceding Article is that of taking-off a model. The corresponding process applied to a real ship is in some respects different; because the measurements have to be made from without, instead of from within.

For the purpose of making such measurements, straight pieces of wood are fixed in suitable positions near the ship, so as to inclose it in a sort of rectangular cage. Some of these are horizontal, or nearly horizontal, and are called *base-boards*; others are vertical, or nearly so, being at right angles to the base-boards, and are called *perpendiculars*. Distances are measured outwards along the upper edges of the base-boards, and upwards along the inner edges of the perpendiculars; and from the ends of those distances, ordinates are measured inwards to the external surface of the ship; and those measurements, being entered in a Table, furnish the means of drawing her calculation-draught.

To find the figure of the stem, a base-board is fixed running out forwards in continuation of the lower edge of the rabbet of the keel, and a perpendicular rising from that base-board a short

distance ahead of the stem. Then by measuring "*distances forward*" along the base-board, with "*ordinates up*," and also "*distances up*" along the perpendicular, with "*ordinates in*," data are obtained for drawing the stem outside the rabbet. By similar operations, the figure of the stern-post outside the rabbet, and that of the stern above the post, are determined; and from these measurements, together with the depth of keel below the rabbet, the outlines of the sheer-plan can be drawn.

To obtain a series of transverse sections, base-boards are fixed running out at right angles to the keel, at suitable intervals apart, such as 10 or 12 feet, and perpendiculars rising from those base-boards so as just to clear the ship's side. Then by measuring "*distances out*" along the base-boards, with "*ordinates up*," and "*distances up*" along the perpendiculars, with "*ordinates in*," data are obtained for constructing the body-plan; from which, together with the sheer-plan, the half-breadth plan can be constructed, thus completing the calculation-draught of the ship; and from the calculation-draught and the thickness of the skin the building-draught may be deduced, if required, as explained in Chapter II., Section I., of this Division.

CORRECTIONS AND ADDENDA TO THE SECOND DIVISION.

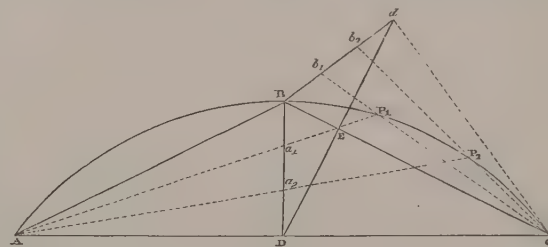
ADDITIONAL NOTE TO ARTICLE 3 OF DIVISION SECOND.—For Mr. John W. Nystrom's method of calculating the ordinates of ships, see the *Journal of the Franklin Institute* for 1863 and 1864, and the *Artizan* for July, 1864.

An ingenious system of approximate formulæ for making various calculations respecting ships is contained in a paper by Mr. J. A. Normand of Havre, entitled "*Mémoire sur l'application de l'algèbre aux calculs des Batiments de Mer.*"

Page 110. first column, line 18 from bottom: for "*bevel*" read "*bevelling*."

ADDENDUM TO ARTICLE 6A.—*Circular Arcs*.—The following is a convenient method of finding points in a circular arc of large radius. In the figure, let A, B, and C be the given points, which are supposed to be equidistant from each other; join A C, A B, B C, and draw B D perpendicular to A C. Divide B D into any number of equal or unequal parts (say equal parts) in the points $a_1, a_2, \&c.$, through D draw D E perpendicular to B C, and produce it to d_1 making $d_1 E$ equal to D E. Join B d_1 and divide B d_1 into the same number of parts in the points, $b_1, b_2, \&c.$, as B D. Join C $b_1, C b_2, C d_1, \&c.$, and also A $a_1, A a_2, \&c.$, and produce A $a_1,$

$\&c.$, to meet the lines, C $b_1, \&c.$ The points P₁, P₂, $\&c.$, in which those lines intersect, are points in the circular arc required; also C d_1 is a tangent at C, and a line



through B, parallel to A C, is a tangent at B. A curve drawn through these points will give the circular arc required.

DIVISION THIRD.

STRENGTH OF MATERIALS, AS APPLIED TO SHIPBUILDING.

CHAPTER I.

ON ELASTICITY AND STRENGTH IN GENERAL.

SECTION I.—GENERAL DEFINITIONS.

1. **ELASTICITY** is the power of a body to resist forces which tend to alter its size and shape. *Fluid* bodies resist alteration of size only. *Solid* bodies, with which we are for the present concerned, resist both alteration of size and alteration of shape, and are said to possess both *elasticity of volume* and *elasticity of figure*. The latter property is that which enables us to build solid materials into *structures*, such as ships, whose utility depends on their preserving certain shapes.

2. **Stress** means at once the intensity of a *load* tending to alter the shape of a solid body, and the intensity of the equal and opposite *resistance* which the body opposes to that load.

3. The word *strain* is commonly used, sometimes in the same sense with the word *stress*, and sometimes to denote the *measure of the alteration of shape* corresponding to a given stress. In precise language it is necessary that each word should have but one meaning; and, therefore, in the present Treatise, the word *strain* will be used only to denote *alteration of shape*.^c

4. The shape of a body when no stress is applied to it, is its *unstrained* or *free* shape. If a previously strained body, upon the removal of the stress, recovers its free shape, it is said to be *perfectly elastic*; if otherwise, *soft*, *plastic*, *ductile*, or *imperfectly elastic*. Absolutely perfect elasticity does not exist in any substance; but for most substances, there are limits of stress below which the imperfection of the elasticity is too small to be of importance in practice.

5. **Set** is that permanent strain or alteration of shape of an imperfectly elastic body which remains after a stress has been removed.

6. **Stiffness** is measured by the intensity of the stress required to produce a certain fixed quantity of strain.

7. **Pliability** is the inverse of stiffness, and is measured by the quantity of the strain produced by a certain fixed stress.

8. **Strength** is the utmost amount of stress which a solid body can bear without breaking. This is the meaning of the word

strength when used alone; and the same meaning is also denoted by *ultimate strength*.

9. **Elastic strength** is the utmost amount of stress which a solid body can bear without *set*. It was formerly believed that the elastic strength of all materials of construction could be definitely determined; but the experiments of Mr. Hodgkinson, and of MM. Chevandier and Wertheim, have shown that for many materials such determination is impossible, owing to the gradual manner in which the set commences.

10. **Proof stress**, or **proof strength**, is the utmost stress which a body can bear without being injured by diminution of its stiffness and strength. It is, therefore, the greatest stress to which any part of a structure intended for after use ought to be subjected by way of *proof* or *trial* of its resistance. The proof strength of a body can be determined by approximation only. It is found experimentally by ascertaining the greatest stress of which the body will bear the repeated or long-continued application without having its ultimate strength diminished.—(See “Report of the Commissioners on the application of Iron to Railway Structures.”) **Proof load** is the load which produces the proof stress.

The proof or testing by experiment of the strength of a piece of material is to be conducted in two different ways, according to the object in view:—

1st. If the piece is to be afterwards used, the testing load must be so limited that it cannot impair the strength of the piece; that is, it must not exceed the proof strength, being from one-half to one-third of the ultimate strength. About double of the intended ordinary working load is in general enough. Care should be taken to avoid shocks, when the testing load approaches near to the proof strength.

2nd. If the piece is to be sacrificed for the sake of ascertaining the ultimate strength of the material, the load is to be increased by slow degrees till the piece breaks.

To find the *proof strength* by experiment, a moderate load is to be applied and removed several times in succession, the *strain* or alteration of figure produced being measured at each application of the load. If that strain does not sensibly increase by repeated applications of the load, the load is within the limits of proof strength. The same experiment is to be repeated with a series of

^c This restriction of the meaning of the word *strain* having been first introduced by the Editor of this Treatise, has since been adopted by Professor William Thomson and some other writers.

gradually increasing loads, until a load is reached whose successive applications produce increasing disfigurements of the piece, when the proof strength will lie between the last load and the last load but one in the series of experiments.

11. *Working stress* is the utmost stress to which it is considered safe to subject a body, during its ordinary use as part of a structure or machine. *Working load* is the load which produces the working stress.

EXAMPLES OF FACTORS OF SAFETY.

MATERIAL.	Breaking Load ÷ Proof Load.	Breaking Load ÷ Working Load.	Proof Load ÷ Working Load.	REMARKS.
Strongest Steel,	1½	—	—	—
Ordinary Steel & Wrought-iron,	2	3	1½	Steady loads.
Do. Do.	2	4 to 6	2 to 3	Moving loads.
Wrought-iron Rivetted Structures,	3	6	2	—
Cast-iron,	2 to 3	3 to 4	About 1½	Steady loads.
Do.	3	6 to 8	2 to 2½	Moving loads.
Timber, average,	3	About 10	3½	—

12. *Factors of safety* are of three kinds:—The ratio in which the breaking load exceeds the proof load; the ratio in which the

breaking load exceeds the working load; and the ratio in which the proof load exceeds the working load. When not otherwise specified, a factor of safety is to be understood in the second of those senses, viz., The ratio in which the breaking load exceeds the working load.

13. *Ultimate strain* is the utmost strain or alteration of shape which a body can bear without breaking, and is proportional to its pliability and ultimate strength jointly. *Proof strain* is the utmost strain which a body can bear without injury, and is proportional to its pliability and proof strength jointly. The strength of a body (ultimate or proof, as the case may be) is obviously proportional to its strain and stiffness jointly.

14. *Spring*, or *resilience*, is the greatest quantity of work or mechanical energy which a body can bear in the form of a blow or shock without injury, and is one-half of the product of the proof strength of the body by its proof strain.

SECTION II.—CLASSIFICATION AND SPECIAL DEFINITIONS.

15. There are as many different kinds of load, stress, strain, stiffness, pliability, strength, and spring, as there are different ways of disfiguring and breaking a body or structure. The more important of these are classified in the following table:—

LOAD AND STRESS.		STRAIN.	STIFFNESS.	PLIABILITY.	WAY OF BREAKING.	STRENGTH.
DIRECT...	1. Pull, or Tension.	Stretching or Extension.	Resistance to Extension.	Direct Extensibility.	Tearing.	Tenacity or resistance to Tearing.
	2. Thrust, or Pressure.	Squeezing or Compression.	Resistance to Compression.	Direct Compressibility.	Crushing.	Resistance to Crushing.
INDIRECT...	3. Shearing or Racking stress.	Racking or Distortion.	Rigidity.	Lateral Pliability.	Shearing, Sliding, or Detrusion.	Resistance to Shearing.
	4. Twisting stress.	Torsion or Twisting.	Resistance to Twisting.	—	Wrenching.	Resistance to Wrenching.
	5. Transverse stress.	Bending.	Transverse Stiffness.	Flexibility.	Breaking across.	Resistance to Breaking across.
	6. Indirect thrust.	Bending with Compression.	—	—	Breaking across.	Resistance to Indirect Crushing.

16. *Combined Stress and Strain*.—The kinds of stress and strain above classified may be *combined* in various ways. For example, a body may be at the same time stretched, twisted, and bent.

17. *Resolved Stress and Strain*.—All the different kinds of indirect stress and strain are capable of being resolved, by means of the principles of statics and geometry, into direct stresses and strains.

18. *Granular and Fibrous Structure*.—When bodies are either of a uniform or a granular texture, like most cast-metals, their strength and stiffness are sensibly the same for all directions of the line in which a direct stress acts. When bodies are of a fibrous structure, like bar-iron, metallic wire, and timber, their strength and stiffness are greater against a stress along than across the fibres.

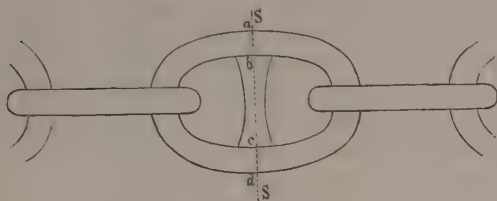
SECTION III.—ON DIRECT STRESS AND STRAIN, AND ON TENACITY.

19. *The Intensity of a Direct Stress*, whether a pull or a thrust, is expressed (in British measures) in *pounds on the square inch*; and is ascertained by dividing the total straining force or load, in pounds, by the area, in square inches, of a cross section of the strained body made by a plane at right angles to the direction of the force. If the strained body is of uniform sectional area (like a rod, bar, or rope) the stress on the square inch is the same at each cross section. If the strained body is of varying sectional area, the stress on the square inch at each cross section is inversely proportional to the area of that section, and is greatest at the section of least area.

20. *Effective Section*.—In determining the sectional area to be employed in calculating a direct stress, it is to be observed that

those parts only of the strained body are to be included which assist in sustaining the stress. For example, let Fig. 1 represent part of a chain cable, stretched by a force whose amount in pounds is given, and let it be required to find the stress at the cross

FIG. 1.



section through the middle of a link. The plane of section *SS* traverses the two sides of the link, *ab* and *cd*, and also the stay *bc*; but as the stay serves merely to prevent the link from collapsing, and does not directly sustain any part of the pull upon the cable, its sectional area is not to be included in computing the stress, which is borne entirely by the two sides of the link *ab*, *cd*.

21. *Even and Uneven Stress.*—In order that the stress may be uniform at every point of a given cross section of a body, it is necessary that the line of action or resultant of the straining force should pass through the *centre of gravity* of the given cross section. If the line of action does not pass through that point, the stress is *unevenly distributed* over the cross section, being greatest at that side of the cross section towards which the line of action deviates. In this case, the stress found by dividing the total straining force by the sectional area, is the *mean stress*, and is equal to the actual stress at the centre of gravity of the cross section. The determination of the actual stress at other points of the cross section involves the principles of the action of *bending forces*, and will be considered in the section relating to that class of forces.

22. The *strain or alteration of shape* produced by a pull or tension, consists in a *stretching or extension* of the body in the direction of the pull, and a *shrinking or contraction* in every direction at right angles to that of the pull. In like manner, the strain or alteration of shape produced by a thrust or pressure consists in a *compression or shortening* of the body in the direction of the thrust, and a *swelling, bulging, or expansion* of the body in every direction at right angles to that of the thrust.

23. The lateral shrinking produced by a pull, and the lateral bulging produced by a thrust, are small compared with the direct extension or compression, and the measurement of their exact amount is difficult in practice. Some of their effects will be afterwards referred to. Meanwhile it is sufficient to state the fact, that if a stretched body be prevented from shrinking laterally, and a compressed body from bulging laterally, the resistance of the former to extension, and the resistance of the latter to compression, are increased.

24. *Measure of Direct Strain.*—The direct extension or compression produced by a pull or thrust is expressed by stating the proportion which the alteration of the length of the strained body, or of some portion of that body, bears to the original length of the body, or of the portion in question, as the case may be. For example, if a bar of iron whose length is 100 inches, be stretched until its length is increased by $\frac{1}{100}$ th of an inch, the extension of that bar is said to be $\frac{0.1 \text{ inch}}{100 \text{ inches}} = \frac{1}{1000}$ or 0.001. To ascertain

the direct extension or compression of a bar or other body, two marks are made upon it at as great a distance apart in the direction of the stress as is conveniently attainable; the distance between those marks is measured in the free state and in the strained state respectively, and the alteration of that distance being divided by the distance in the free state, gives the amount of the *direct strain* or proportional alteration of length.

25. *Direct Pliability.*—The amount of direct strain produced by each pound on the square inch of direct stress, is called the *direct extensibility* or *compressibility* of the body, as the case may be; and both these terms may conveniently be comprehended under the single term *direct pliability*. Thus, if it be found by experiment that a pull of 28,000 pounds on the square inch lengthens a bar of iron by $\frac{1}{280000}$ of its original length, the amount of strain for each

pound of stress on the square inch is $\frac{1}{28000000} = 0.0000000357$,

and this fraction is called the *extensibility* of the iron under experiment. In most substances the extensibility and compressibility are nearly uniform and equal to each other, for stresses not exceeding the *proof stress*; and for practical purposes they may be treated in calculation as exactly uniform and equal to each other up to that limit. In fact, a marked and sudden increase of pliability is one experimental test that the limit of proof strength has been exceeded; for it indicates that the stress is great enough to weaken the material.

26. *Modulus of Elasticity.*—The reciprocal of the direct pliability, as determined from experiment with stresses not exceeding the proof strength, is called conventionally the *modulus of elasticity*. For example, the modulus of elasticity of the iron referred to in the preceding article, is 28,000,000 lb. per square inch. It is to be observed, that the modulus of elasticity, as thus defined, is the modulus of one particular kind of elastic force only, viz., that which resists direct extension and compression; nevertheless, the general term "Modulus of Elasticity" is retained in compliance with custom. The values of this quantity for various substances are given in the annexed tables of the properties of materials in the column headed E. Where the substances are fibrous, the values of the modulus given are to be understood to apply to the *direction of the fibres*; for in a direction across the fibres, the resistance to direct stress is less than that along the fibres. The elasticity in a direction *across the grain* has been determined for a few kinds of timber only; its values in some cases are given in an additional table. In some cases, the modulus of elasticity, as given in the principal tables, has been deduced by an indirect process from the results of experiments on bending by a transverse load. These cases will be further referred to under the head of Bending.

27. Some of the *uses of the modulus of elasticity in computation* are as follows:—First, *To find the direct pliability of the substance per pound of stress on the square inch*—Take the reciprocal of the modulus of elasticity (which may be done either by division or by a table of reciprocals). For example, the modulus of elasticity of Norway red pine is 1,458,000 lb. per square inch; hence the direct pliability of that timber is $\frac{1}{1458000} = 0.0000006859$ nearly, per pound of stress on the square inch.

Secondly, *To find the stress in pounds per square inch required to produce a given direct strain*—Multiply the strain by the modulus of elasticity. For example, to find the pull per square inch required to elongate a bar of the iron referred to in articles

25 and 26 by one-thousandth part of its length, we have—

$$\frac{\text{Modulus.}}{28000000} \times \frac{\text{Strain.}}{.001} = \frac{\text{Stress.}}{28000}.$$

Thirdly, *To find the direct strain produced by a given direct stress—Divide the stress by the modulus of elasticity.* Thus, in

the last example we have $\frac{\text{stress } 28000}{\text{modulus } 28000000} = \text{strain } \frac{1}{1000}$, being the proportionate elongation of the bar.

28. *Tenacity.*—If the pull upon a bar be gradually and continually increased, a limit is at length reached at which the material ceases to be able to resist the stress; and then the bar is torn asunder. The utmost pull per square inch of cross section which a bar of a given material sustains just before being torn asunder, is called the *tenacity*, *ultimate tenacity*, or *tearing force* of the material, and sometimes the *direct cohesion*. It is denoted in the tables of this article by the symbol F_t . It is usually expressed in *pounds per square inch*; but sometimes also in *tons per square inch*.

29. The *proof tenacity*, or limit of the pull per square inch of section which a bar of a given material can bear without injury, is indicated in experimenting, especially with ductile materials, either by an increase in the extensibility of the material, when a pull exceeding that limit is applied, or by an increase of extension at each time that such pull is re-applied after having been removed; showing that the stress, although not sufficient to tear the bar asunder at once, would ultimately do so if its application were often enough repeated. The proof tenacity, generally speaking, approaches nearest to the ultimate tenacity in the stiffest and strongest materials—a fact illustrated by the table annexed to article 12. The ordinary *working pull* is from one-half to two-thirds of the proof tenacity with a steady load, and from one-third to one-half with a moving load.

30. *Tenacity of Rivetted Joints.*—It is important in practice to know, not only the tenacity of single bars of a given material, but also that of articles made by putting materials together in various ways. Of this kind are the joints of plates of iron rivetted in different ways, and the cells, ribs, and other combinations of plates and bars which occur in iron beams and framework. The results of experiments, chiefly by Mr. Fairbairn, on such structures prove, that when the rivet-holes are deducted, the tenacity per square inch of solid metal left between the rivet-holes is in double-rivetted joints as great as in the continuous plate, and in single-rivetted joints about four-fifths of that amount, as the following table shows:—

RELATIVE TENACITIES.*

	Rivet-holes deducted.	Rivet-holes included.
Continuous plate,.....	100	100
Double-rivetted joint,.....	100	70
Single-rivetted joint,.....	79	56

The strength of the rivets themselves will be further considered under the head of Resistance to Shearing.

31. *Tenacity of Ropes.*—The tenacity of ropes of iron wire and of hemp is given in the table in pounds per square inch of section. It is sometimes more convenient, however, to refer the tenacity of wire ropes to their weight, and that of hempen cables to the square of their girth; and for that purpose the following data may be taken:—

IRON WIRE ROPES.—Tenacity for each pound that one fathom of the rope weighs.

Ultimate or tearing force, 2 tons = 4480 lb., = 4480 fathoms of the rope itself.

Proof stress, 1 ton = 2240 lb. = 2240 fathoms of the rope.

Working pull, $\frac{1}{2}$ ton = 747 lb. = 747 fathoms of the rope.

HEMPEN CABLES.—Ultimate tenacity = square of the girth in inches \times 448 lb. or $\frac{1}{2}$ ton.

32. *Hollow Cylinders.*—I. To compute the intensity of the stress on the material of a *thin hollow cylinder* with a pressure from within tending to burst it, such as a boiler or a pipe—*Multiply the intensity of the effective pressure from within by the number of times that the inside radius is greater than the thickness of metal.*

II. To find of what thickness the metal of the cylinder ought to be, in order that the tension tending to burst it may not exceed a given safe limit, with a given effective pressure from within—*Make the thickness of metal less than the inside radius, in the same proportion that the effective pressure is less than the given safe tension.*

(In algebraical symbols, let r be the inside radius, m the thickness of metal, p the effective pressure, f the tension; then—

$$f = \frac{p r}{m}; \quad m = \frac{p r}{f}.$$

These rules, which are founded on the supposition that the tension is uniformly distributed throughout the thickness of the metal, are near enough to the truth for practical purposes when the thickness of the cylinder is not more than about one-tenth of its radius. But for greater thicknesses, it becomes necessary to take into account the fact, that the inner layers of the metal are more severely stretched than the outer layers; and the results are as follows:—

III. To find the ratio in which the greatest tension of the metal is greater than the effective pressure from within—*Divide the sum of the squares of the internal and external radii by the difference of those squares:—*

IV. To find the ratio of the external to the internal radius when the intensities of the effective pressure and of the greatest tension are given—*Divide the sum of those intensities by their difference, and extract the square root of the quotient.*

(In algebraical symbols, let p be the effective pressure, f the greatest tension, R the external and r the internal radius; then—

$$\frac{p}{f} = \frac{R^2 - r^2}{R^2 + r^2}; \quad \text{and} \\ \frac{R}{r} = \sqrt{\left(\frac{f + p}{f - p}\right)}.$$

33. *Hollow Spheres.*—A *thin* spherical shell is twice as strong to resist bursting as a cylindrical shell of the same radius and thickness; hence, to resist a given internal pressure, a thin spherical shell should be of one-half the thickness required for a cylindrical shell of the same radius with the same pressure.

When a cylindrical shell has spherical ends, there are practical reasons for making the spherical parts of the same radius and thickness with the cylindrical part, notwithstanding that they are thereby made twice as strong.

The following are the rules applicable to a *thick* hollow sphere, pressed from within:—

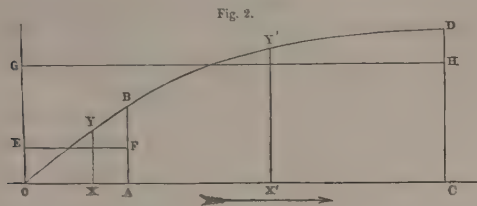
I. *To find the ratio of the greatest safe internal effective pressure to the greatest safe tension on the material, in a hollow sphere of given radii—Divide twice the difference of the cubes of the radii by the sum of the cube of the external radius and twice the cube of the internal radius.*

II. *To find the ratio of the external to the internal radius, when*

* Fairbairn's Useful Information.

the greatest safe pressure and greatest safe tension are given. Divide twice the sum of the greatest safe tension and greatest safe pressure, by a divisor found by subtracting the greatest safe pressure from twice the greatest safe tension; extract the cube root of the quotient.

34. *Work of Stretching and Tearing—Resilience of a Tie.*—In Fig. 2, let O represent the position of the right-hand end of a horizontal bar in its free condition, the other end being



fixed somewhere to the left of the figure. When a gradually increasing stretching load is applied to the bar, let its right-hand end move from O towards C, so that a scale of distances, or *abscissæ* (see Division I, Article 5), measured from O towards C, shall represent the successive *elongations* of the bar. Let *ordinates*, at right angles to OC, represent, according to a suitable scale, the *loads* required to produce those elongations; so that, for example, XY shall represent the load which produces the elongation OX, X'Y' the load which produces the elongation OX', &c.

Let OC represent the elongation immediately before breaking, and CD the breaking load; AB the *proof load*, and OA the elongation produced by it.

The heads of the ordinates will lie in a line OYBY'D of some definite figure. When the load does not exceed the proof load, the elongation bears a nearly constant proportion to the load (Article 25), viz.:—

$$\frac{\text{Load}}{\text{Sectional area}} = \frac{\text{Elongation}}{\text{Original length}} \times \text{Modulus of elasticity};$$

so that from O to B the line in question is nearly straight. When the load exceeds the proof load, the elongation increases more rapidly than the load, in a continually increasing proportion, until the bar is torn asunder; hence from B to D the line presents a curve, concave towards the axis of abscissæ or scale of elongations, OC.

The *Mechanical Work* done in producing a given elongation, OX', is represented by the area OX'Y'.

The area OAB represents the work done in producing the *proof elongation*, and this is what is called the *resilience* of the bar (Article 14). Inasmuch as OB is nearly a straight line, the area OAB is nearly that of a triangle—that is, half the product of the base and height; and therefore,

I. *The Resilience of a tie-bar is (sensibly) half the product of the proof load and proof elongation.*

The proof load is equal to the *proof stress* multiplied by the sectional area of the bar; the proof elongation is equal to the length of the bar, multiplied by the proof stress, and divided by the modulus of elasticity; consequently,

$$\text{II. Resilience} = \frac{(\text{Proof stress})^2}{\text{Modulus of elasticity}} \times \frac{\text{Area} \times \text{Length}}{2}.$$

[Or, in algebraical symbols, let f be the proof stress, E the modulus of elasticity, S the sectional area of the bar, and L its length; then—

$$\text{Resilience} = \frac{f^2}{E} \cdot \frac{SL}{2}.]$$

The square of the proof stress, divided by the modulus of elasticity ($\frac{f^2}{E}$), is called the *Modulus of Resilience*; and when the proof stress and modulus of resilience are expressed in *pounds on the square inch*, it represents, in *foot-pounds of mechanical energy*, the *resilience of a tie-bar one inch square and two feet long*.

The following are examples:—

Material.	Proof Stress.	Modulus of Elasticity.	Modulus of Resilience.
Cast Iron,.....	5,500	17,000,000	1.8
Bar Iron,.....	20,000	28,000,000	14.3
Iron Wire,.....	30,000	25,000,000	36.0
Steel,.....	36,000	28,000,000	46.3

But these are given rather to illustrate the rule than to offer precise results, for which data are wanting.

The area OCD represents the work done in *tearing the bar asunder*. Owing to the form of the curve between B and D, that area is more, and often considerably more, than one-half of the rectangle of the base and height; so that the work of tearing is more than one-half of the product of the breaking load and ultimate elongation. In some of Mr. Kirkaldy's experiments on iron, indeed, it would seem to have been as much as *four-fifths* of that product.

35. *Effect of a Sudden Load.—Dead and Live Load.*—The load which, if suddenly applied, produces a given elongation, is found by *dividing the work of elongation by the elongation*, and is equal to the mean of all the values of the gradually increasing load by which the same elongation is produced. For example, the sudden load required in order to produce the elongation OX' is expressed by

$$\frac{\text{Area, OX'Y'}}{\text{Base, OX'}}$$

and is represented by the height of a rectangle equal in area to OX'Y', and standing on the same base.

Thus, the sudden load required in order to produce the proof strain, is represented by the height OE of a rectangle, OEFA, whose area is equal to OAB; and OAB being sensibly a right-lined triangle, we have OE = $\frac{1}{2}$ AB, and the *sudden load equal to one-half of the proof load*.

Hence the rule, often followed in practice, that the factor of safety for a movable or "*live*" load on a structure should be twice as great as the factor of safety for a fixed or "*dead*" load. For example, if the factor of safety for a dead load on any iron structure is 3, it should be 6 for a live load; so that the ultimate strength of an iron structure should not be less than—

$$(\text{Dead Load} \times 3) + (\text{Live Load} \times 6).$$

The suddenly-applied load required in order to *break* the bar is represented by OG, being the height of a rectangle OGHG, of area equal to ODC. For the reason explained in the preceding Article, that height is more than one-half of CD; so that the sudden breaking load is more than one-half of the gradually applied breaking load. In the experiments of Mr. Kirkaldy already referred to, it was about four-fifths.

The principles explained in this Article are applicable to every way in which a piece of material can be loaded, as well as to tension.

SECTION IV.—OF CRUSHING.

36. *Compression and Crushing in General.*—Resistance to *Longitudinal Compression*, when the proof stress is not exceeded, is sensibly equal to the resistance to stretching, and is expressed by the same modulus. When that limit is exceeded, it becomes irregular.

In framework generally, a piece of material which is subjected to longitudinal compression, is called a *strut*, and should it happen to stand upright, a *pillar* or *column*. Pieces in the framework of a ship which act as pillars are called *stanchions*.

Crushing, or breaking by compression, is not a simple phenomenon like tearing, but is more or less complex and varied, according to the nature of the substance.

There are two principal ways in which the crushing of a piece of material may take place, distinguished as *direct crushing*, and *crushing by cross-breaking*.

37. *Direct crushing* takes place when the piece along which the thrust acts is not so long in proportion to its diameter as to have a sensible tendency to give way by bending sideways. Such is the condition of—

Pillars, rods, and struts of cast iron, in which the length is not more than five times the diameter, approximately;

Pillars, rods, and struts of wrought iron, in which the length is not more than ten times the diameter, approximately;

Pillars, rods, and struts of dry timber, in which the length is not more than about four times the diameter.

The *direct crushing load* of a piece of material, when that load is uniformly distributed, is equal to the area of its transverse section in square inches, multiplied by the modulus of resistance of the material to crushing in lbs. on the square inch.

If the load is not uniformly distributed over the transverse section of the piece, the strength of the piece is diminished in the same ratio in which the mean intensity of the stress is less than the maximum intensity.

This case will be further considered in a later section.

The modulus of resistance to direct crushing, as the annexed Tables show, often differs considerably from the tenacity. The nature and amount of the difference depend mainly on the mode in which the crushing takes place. Modes of direct crushing may be classed as follows:—

I. *Crushing by splitting* into a number of nearly prismatic fragments, separated by smooth surfaces whose general direction is nearly parallel to the direction of the load, is characteristic of hard homogeneous substances of a glassy texture, such as vitrified bricks.

II. *Crushing by shearing or sliding* of portions of the block along oblique surfaces of separation is characteristic of substances of a granular texture, like cast iron, and most kinds of stone and brick. Sometimes the sliding takes place at a single plane surface; sometimes two cones or pyramids are formed, point to point, which are forced towards each other, and split or drive outwards a number of wedges surrounding them.

The surfaces of shearing make an angle with the direction of the crushing force, which Mr. Hodgkinson (who first fully investigated those phenomena) found to have values depending on the kind and quality of material. For different qualities of cast iron, for example, that angle ranges from 42° to 32° . The greatest intensity of shearing stress is on a plane making an angle of 45° with the direction of the crushing force; and the deviation of the plane of shearing from that angle shows that the resistance to shearing is not purely a cohesive force, independent of the normal pressure at the plane of shearing, but consists partly of a force analogous to friction, increasing with the intensity of the normal pressure.

Mr. Hodgkinson considered that in order to determine the true resistance of substances to direct crushing, experiments should be made on blocks in which the proportion of length to diameter is not less than that of 3 to 2, in order that the material may be free to divide itself by shearing. When a block which is shorter in proportion to its diameter is crushed, the friction of the flat surfaces between which it is crushed has a perceptible effect in *holding its parts together*, so as to resist their separation by shearing; and thus the apparent strength of the substance is increased beyond its real strength.

In all substances which are crushed by splitting and by shearing, the resistance to crushing considerably exceeds the tenacity, as the Tables of strength show. The resistance of cast iron to crushing, for example, was found by Mr. Hodgkinson to be somewhat more than six times its tenacity.

III. *Crushing by bulging*, or lateral swelling and spreading of the block which is crushed, is characteristic of ductile and tough materials, such as wrought iron. Owing to the gradual manner in which materials of this nature give way to a crushing force, it is difficult to determine their resistance to that force exactly. That resistance is in general less, and sometimes considerably less, than the tenacity. In wrought iron, the resistance to the direct crushing of short blocks, as nearly as it can be ascertained, is from $\frac{2}{3}$ to $\frac{4}{5}$ of the tenacity.

IV. *Crushing by buckling or crippling* is characteristic of fibrous substances, under the action of a thrust along the fibres. It consists in a lateral bending and wrinkling of the fibres, sometimes accompanied by a splitting of them asunder. It takes place in timber, in plates of wrought iron, and in bars longer than those which give way by bulging. The resistance of fibrous substances to crushing is in general considerably less than their tenacity, especially where the lateral adhesion of the fibres to each other is weak compared with their tenacity. The resistance of most kinds of timber to crushing, when dry, is from $\frac{1}{2}$ to $\frac{2}{3}$ of the tenacity. Moisture in the timber weakens the lateral adhesion of the fibres, and reduces the resistance to crushing to about one-half of its amount in the dry state.

38. *Crushing by cross-breaking* is the mode of fracture of columns and struts in which the length greatly exceeds the diameter. Under the breaking load, they yield sideways, and are broken across like beams under a transverse load.

The most convenient form of rule for calculating the resistance of long struts and pillars to crushing by cross-breaking, with a degree of accuracy sufficient for practical purposes, may be expressed thus:—

I. *The ultimate resistance per unit of sectional area is found by dividing a constant modulus by unity, added to the product of the square of the ratio in which the length of the piece exceeds its least outside diameter, into a constant multiplier.* In other words, divide the length of the piece by its least outside diameter; square the quotient, and multiply the square by a suitable constant multiplier; to the product add unity; divide the proper constant modulus by the sum; the quotient will be the resistance to crushing per unit of sectional area of the piece.

[In algebraical symbols,

Let P be the crushing load of a long rod or pillar in lbs.;

S the sectional area of material in it, in square inches;

l , its length,

h , its least external diameter, } both in the same units of measure.

Then, approximately—

$$P = \frac{fS}{1 + a \cdot \frac{l^2}{h^2}}$$

The modulus (f) depends on the nature of the material, and is nearly, though not quite, equal to the modulus of the resistance of short blocks to direct crushing.

The multiplier in the divisor (a) depends on the nature of the material, the mode of fixing of the piece, and its form of cross-section; being greater in proportion as the form of cross-section is more flexible.

The following Table gives some values of those constants for pillars or struts *fixed at both ends*. It is to be observed, however, with respect to the multiplier (a), that three only of its values—viz. those marked †—have been deduced directly from experiment, the others having been inferred from those three by the probable supposition that they are proportional to the flexibility.

Material.	Modulus f , Lbs. on the square inch.	Multiplier a for Struts fixed at both ends.	Figure of Strut.
Cast Iron,.....	80,000	$\frac{1}{3000}^\dagger$	Hollow cylinder.
Wrought Iron,.....	36,000	$\frac{1}{3000}^\dagger$	Solid rectangle.
"	"	$\frac{1}{3000}$	Thin square tube, or cell.
"	"	$\frac{1}{3000}$	Solid cylinder.
"	"	$\frac{1}{3000}$	Thin cylindrical tube.
"	"	$\frac{1}{3000}$	Angle iron.
"	"	$\frac{1}{3000}$	Cross-shaped section.
Timber, average.....	6,500	$\frac{1}{3000}^\dagger$	Solid rectangle.

II. For struts and pillars *jointed at both ends*, multiply the value of a given in the Table by 4.

III. For struts and pillars *fixed at one end and jointed at the other*, multiply the value of a given in the Table by 2.

In using the preceding rules for the purpose of finding the dimensions of a strut to bear a given thrust, the proportion of the length to the least outside diameter is usually fixed beforehand. This enables the ultimate resistance per square inch of section to be computed, by which the intended breaking load is to be divided, in order to find the required sectional area.

In designing columns, it is advisable to use large factors of safety; say, for cast iron, 8; for wrought iron, 6; for timber, 10.

39. *Collapsing*.—When a thin hollow cylinder is pressed from without, it gives way by *collapsing*, under a pressure whose intensity has been found by Mr. Fairbairn (*Philos. Trans.*, 1858)* to vary nearly according to the following laws:—

Inversely as the length;

Inversely as the diameter;

Directly as a function of the thickness, which is very nearly the power whose index is 2.19; but which for ordinary practical purposes may be treated as sensibly equal to the *square* of the thickness.

The following rule gives approximately the *collapsing pressure* in lbs. on the square inch of a plate-iron flue with butt-joints, whose length, diameter, and thickness are all expressed in the same units of measure; say, all in inches, or all in feet:—

I. *Divide the square of the thickness by the length and by the diameter, and multiply the quotient by 9,672,000.*

When the thickness and diameter are expressed in inches, and the length in feet, the multiplier becomes one-twelfth of the above, or 806,000.

When the thickness is expressed in inches, and the length and diameter in feet, the multiplier becomes 67,000 nearly.

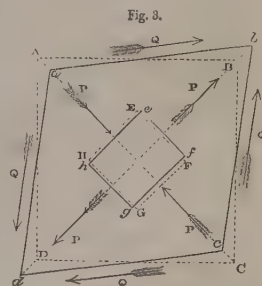
When a tube is strengthened by rivetting round it rings of T-iron at equal distances apart, the resistance to collapsing is that corresponding to the length *from ring to ring*.

II. The *resistance of an elliptical tube to collapsing*, is nearly the same with that of a circular tube whose curvature is the same with that of the flattest part of the ellipse. To find the diameter of that circular tube, *divide the square of the greatest diameter of the ellipse by its least diameter*.

SECTION V.—OF RACKING AND SHEARING.

40. *Racking or Distortion* is that kind of disfigurement of a piece of material which takes place when originally rectangular parts of it become oblique-angled, without alteration of volume.

For example, in Fig. 3, the dotted outline, $ABCD$, represents the original or free figure of a square block, which is racked or distorted into the rhombic figure, $abed$, without alteration of size; the originally right angles, B and D , being changed into the acute angles, b and d , and the originally right angles, A and C , into the obtuse angles, a and c .



It is obvious that the same alteration of figure might be otherwise described, by saying that the diagonal, \overline{BD} , is stretched to the length, \overline{bd} , and the other diagonal, \overline{AC} , shortened to the length \overline{ac} ; the two diagonals continuing to be at right angles to each other, and the one being lengthened in the same proportion that the other is shortened, so that the area of the figure continues unchanged; viz.:—

$$\frac{1}{2} \cdot \overline{ac} \times \overline{bd} = \frac{1}{2} \cdot \overline{AC} \times \overline{BD}.$$

This kind of strain is expressed as a quantity, by stating, *in circular measure* (Div. I., Article 30, page 15), the difference between any one of the altered angles and a right angle. In practice, that difference is always a very small fraction, and then it is sensibly equal to *double the proportionate alteration of length undergone by the diagonals*; that is to say—

$$\text{Distortion} = 2 \cdot \frac{\overline{AC} - \overline{ac}}{\overline{AC}} = 2 \cdot \frac{\overline{bd} - \overline{BD}}{\overline{BD}} \text{ sensibly.}$$

Let $EFGH$ represent an originally square particle of the block, having its faces perpendicular to the diagonals of the block. When the block is racked, it is evident that this particle becomes of an oblong figure, like $efgh$; being stretched in the direction of BD , and compressed in an equal proportion in the direction of AC .

41. *Racking or Shearing Stress* is that kind of stress which produces and accompanies distortion. It is a force *tangential* or parallel to the surfaces of the particles between which it acts. Thus, to produce the distortion represented in Fig. 3, a racking stress must be applied to each of the four faces of the block, as shown by the arrows marked Q .

The two following principles regarding racking stress, are demonstrated in treatises on the elasticity of solids:—

I. The racking stress at the two pairs of faces of a distorted particle is of equal intensity;

II. Every racking stress on a particle is equivalent to the combination of a tension and a thrust of the same intensity, acting diagonally, or at angles of 45° , as regards the racking stress.

* See also *Useful Information for Engineers*. Second series, 1860.

For example, the racking stress, of a certain number of pounds on the square inch of surface, represented by the arrows marked Q, is equivalent to the combination of a tensile stress, of the same number of pounds on the square inch, acting parallel to the diagonal, BD, and a compressive stress, of the same number of pounds on the square inch, acting along the diagonal, AC.

42. By *Modulus of Rigidity* is meant a coefficient bearing the same relation to racking strain that the modulus of direct elasticity bears to direct extension and compression; that is to say, if the distortion of a particle is expressed as in Article 40, and the racking stress as in Article 41, we have—

Racking stress = Distortion \times Modulus of rigidity, and

$$\text{Distortion} = \frac{\text{Racking stress}}{\text{Modulus of rigidity}}.$$

The modulus of rigidity has been ascertained for very few substances. The following are some of its values:—

METALS.	Modulus of Rigidity, Lbs. on the square inch.
Brass Wire,.....	5,330,000
Copper,.....	6,200,000
Iron, Cast,.....	2,850,000
Iron, Wrought, from,.....	8,500,000
" " to,.....	10,800,000
TIMBER (Plane of Distortion parallel to the fibres).	
Red Pine, from,.....	62,000
" to,.....	116,000
Elm,.....	76,000
Ash,.....	76,000
Oak (from North of France),.....	82,000

The rigidity bears various proportions to the direct elasticity in different substances. It is proportionally greatest in the hardest substances, and least in those which are soft or gelatinous.

43. *Resistance to Shearing*.—A piece of material (such as the block in Fig. 3), subjected to a racking load which exceeds its strength, may break in one or other of three ways; by tearing along the stretched diagonal, *b d*, by bending or buckling along the compressed diagonal, *a c*, or by *shearing* or sliding of one part upon another, in a plane parallel to one or other of the racking stresses, Q.

When the body gives way by shearing, the *Modulus of Resistance to Shearing* is the number of units of shearing stress on each unit of area of the surface at which shearing takes place. It is given in the Tables for various substances, in lbs. on the square inch.

In tough compact metals, like plate-iron and rivet-iron, the resistance to shearing is nearly equal to the tenacity; in fibrous materials, such as timber, it is nearly equal to the tenacity across the grain; in granular materials, like cast iron, whose resistance to direct crushing is greater than the tenacity, the resistance to shearing is in general somewhat greater than the tenacity.

In ships and other structures, many cases occur in which the principal pieces, such as plates, bars, or beams, being themselves subjected to a direct pull, are connected with each other at their joints by fastenings, such as rivets, bolts, pins, keys, or screws, which are under the action of a shearing force, tending to make them give way by the sliding of one part over another.

In order that the resistance of a fastening exposed to a shearing force may be the greatest possible, the stress should be uniformly distributed; and to insure uniform distribution of the stress, it is necessary that the rivet or other fastening should fit so tight in its hole or socket, that the friction at its surface may be at least of equal intensity to the shearing stress.

This condition is seldom completely fulfilled in practice; and it is therefore necessary to make allowance in calculation for the weakening effect of slackness, which may be thus estimated:—

FORM OF PIN, KEY, OR BOLT.	Strength reduced to
Rectangular,.....	$\frac{3}{4}$ } of that of a tight fastening.
Cylindrical,.....	$\frac{2}{3}$ }

The effects of racking stress varying in intensity will be considered under the head of "Resistance to Bending," which is in general accompanied by such a stress; and the effects of racking stress varying in direction as well as in intensity, under the head of "Resistance to Twisting."

SECTION VI.—OF BENDING AND CROSS-BREAKING.

44. *Actions of a Transverse Load—Racking Force—Bending Moment*.—In general, a transverse load applied to a beam (or to any structure, such as a ship, which may be considered as analogous to a beam), produces both racking and bending at each cross-section of the beam. The *Racking Force*, at a given cross-section of the beam, is the *resultant* (see Division I., Article 59, page 25) of all the forces which act at right angles to the length of the beam, upon one of the two parts into which that cross-section divides it. The *Bending Moment*, at a given cross-section of the beam, is the *resultant moment* (see Division First, Article 58, page 25) of the same set of forces.

Loads upon beams and racking forces are stated in British measures, either in pounds, hundredweights, or tons; lengths of beams, in feet, or in inches; and according to the units of load and length employed, the unit of bending moment is a *foot-pound*, an *inch-pound*, a *foot-hundredweight*, an *inch-hundredweight*, a *foot-ton*, or an *inch-ton*, as the case may be.

As the transverse dimensions of beams are expressed in inches, and their moduli of strength in pounds on the square inch, the most convenient British units are, the pound, the inch, and the inch-pound.

The following is a comparison of different British *units of bending moment*—

Inch-lbs.	
12 =	1 Ft.-lb.
112 =	9½ = 1 Inch-cwt.
1344 =	112 = 12 = 1 Foot-cwt.
2240 =	186½ = 20 = 1½ = 1 Inch-ton.
26880 =	2240 = 240 = 20 = 12 = 1 Foot-ton.

When loads are expressed in *cubic feet of sea-water*, they may be converted into pounds by multiplying them by 64.

As to metrical units of load and moment, see Division First, Articles 45 and 46, pages 21, 22.

The racking force and bending moment are resisted and balanced by the stress of the particles at the cross-section in question, according to principles which will be stated in the two following Articles. The present Article relates specially to the mode of computing the racking forces and bending moments produced at given cross-sections of a beam, by a given load distributed in a given way. The following are the cases which oftenest occur in practice:—

CASE I.—*Beam acted upon by three parallel forces*. In Figs. 4 and 5, AB represents the axis of a beam acted upon by three parallel forces at right angles to its length, which three forces act in one plane, and balance each other, and are represented by the arrows at A, B, and G. The beam may have various different positions—horizontal, inclined, or vertical: the two most common positions being those represented in the figures.

Fig. 4 represents a horizontal beam supported at the ends, A and B, and loaded with a weight at an intermediate point, G: the arrows at A and B represent the upward supporting pressures exerted on the beam by the two props. The distance AB is called the *span* of the beam.

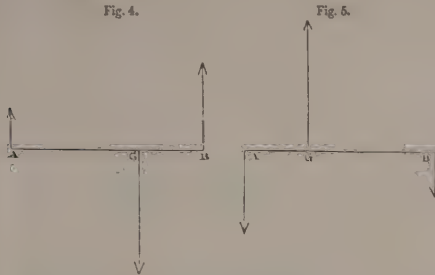


Fig. 5 represents a horizontal beam loaded with a weight at one end, A, supported by a prop at an intermediate point, G, and held down by a suitable fastening at the other end, B.

In every case the relations amongst those three forces are regulated by principles which are particular instances of the general principles stated in Article 59 of Division I., page 25; viz. :—

Because of the balance of forces, *the middle force (at G) is opposite in direction to the other two forces, and equal to their sum.*

Because of the balance of moments, *each of the three forces is proportional to the distance between the lines of action of the other two.*

Those principles lead to the following preliminary rules, to be used when one only of the three forces is given, and the others are to be computed :—

RULE A.—The beam in Fig. 4 being supported at A and B, and loaded at G, and the load at G being given, to find the supporting forces at A and B :—

$$\text{Supporting force at A} = \text{Load at G} \times \frac{\overline{GB}}{\overline{AB}};$$

$$\text{Supporting force at B} = \text{Load at G} \times \frac{\overline{GA}}{\overline{AB}}.$$

RULE B.—The beam in Fig. 5 being loaded at A, supported at G, and held down at B, and the load at A being given, to find the forces at G and B :—

$$\text{Supporting force at G} = \text{Load at A} \times \frac{\overline{AB}}{\overline{GB}};$$

$$\text{Holding-down force at B} = \text{Load at A} \times \frac{\overline{GA}}{\overline{GB}}.$$

The three forces being now known, the following rules serve to find the racking force and bending moment at any given cross-section :—

RULE C.—The *racking actions* upon the two parts into which the point G divides the span are opposite in direction; and the racking force at any given cross-section is equal to the force applied at that end of the beam between which and the point, G, the given cross-section is situated; that is to say, at any cross-section between A and G, the racking force is equal to the force applied at A; and at any cross-section between G and B, the racking force is equal to the force applied at B.

RULE D.—The *bending action* tends to make the beam become convex in the direction towards which the force at the intermediate point, G, acts; and the bending moment at any given cross-section is equal to the force applied at the end of the beam, between which

and the point, G, the given cross-section is situated, multiplied by the distance of that end from the given cross-section; that is to say, at any cross-section between A and G, the bending moment is—

$$\text{Force at A} \times \text{distance from A};$$

and at any cross-section between G and B, the bending moment is—

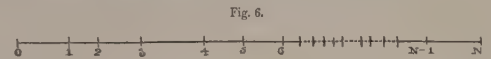
$$\text{Force at B} \times \text{distance from B}.$$

RULE D².—The *greatest bending moment* evidently occurs at the point, G, where the direction of the racking action changes; and its value may be expressed in either of the three following ways :—

$$\text{Force at A} \times \overline{AG} = \text{Force at B} \times \overline{GB}$$

$$= \text{Force at G} \times \frac{\overline{AG} \cdot \overline{GB}}{\overline{AB}}$$

CASE II.—*Beam under several Parallel Forces.*—Let ON, in Fig. 6, represent the axis of a beam, perpendicular to which forces



are applied at a series of detached points; viz., the ends of the beam marked O and N, and intermediate points, marked 1, 2, 3, &c., up to N—1.

In many of the cases which occur in practice, the beam is supported at two points, and loaded at all the others (the supported points being usually the endmost points, O and N); and the forces applied at the loaded points are given, leaving the two supporting forces to be found by a preliminary calculation. To make that preliminary calculation, the position and magnitude of the *resultant* of the load are, in the first place, to be found by the principles of Division First, Article 59, page 25; that is to say :—

RULE E.—For the *magnitude of the resultant load*, add together its components; and for its *distance from O*, multiply each component by the distance of that component from O, and divide the sum of the products by the resultant load. Then conceiving, for the moment, that instead of the real load, there is substituted its resultant, concentrated at one point, like G in Fig. 4, the *two supporting forces* are to be found by Rule A.

In other cases, known supporting forces may act at more than two points, and loading forces at the others; subject to the condition, that the sum of the supporting forces must be equal to the sum of the loading forces, and the resultant moment of the supporting forces relatively to any transverse axis (such, for example, as an axis traversing O), shall be equal and opposite to the resultant moment of the loading forces relatively to the same axis.

The applied forces being completely known, the next step is to find the series of *racking forces* which act on the several divisions of the span of the beam, $\overline{O1}$, $\overline{12}$, $\overline{23}$, &c. This is done by the following Rule :—

RULE F.—The *racking force on any division of the beam is equal to the resultant of the forces which are applied to the beam at points between that division and either end of the span.* In order to find that resultant, the applied forces must be distinguished into positive and negative, according to their direction; and then by their successive addition and subtraction, the series of racking forces are found, one after another. For example, in a beam supported at O and N, and loaded at intermediate points, the series of racking forces is found thus—

The racking force on the division $\overline{O1}$ is equal to the supporting force at O.

Subtract the load at the point 1; the remainder will be the racking force on the division 1 2;

From that force subtract the load at the point 2; the remainder will be the racking force on the division 2 3; and so on.

In pursuing this process, a point is at length reached where the load is greater than the racking force on the preceding division, so that their difference, being the racking force on the following division, is *negative*. This shows, that at the point in question the racking action reverses its direction, so that the point may be called the *point of reversed racking*. The remaining terms of the series of racking forces are found by adding instead of subtracting the successive loads, it being borne in mind, that the forces so found are all negative—that is, contrary in direction to the positive racking forces.

As a check upon the accuracy of the calculations, the racking force on the last division ($N-1$) (N), ought to be equal to the supporting force at N.

Finally, the series of bending moments at the points 1, 2, 3, &c., is to be found as follows:—

RULE G.—Multiply each racking force by the length of the division on which it acts; the bending moment at any given loaded point, is equal to the algebraical sum of the products corresponding to the divisions which lie between that point and either end of the beam.

For example, the bending moment at the point O is = 0;

The bending moment at the point 1 is equal to the product of the shearing force on the division O 1, by the length of that division;

To that moment, add the product of the shearing force on the division 1 2 by the length of that division; the sum will be the bending moment at the point 2;

To that moment, add the corresponding product for the division 2 3; the sum will be the bending moment at the point 3; and so on.

At the *point of reversed racking*, the bending moment reaches a maximum; for the products belonging to the following divisions of the span, being negative, are to be *subtracted* successively instead of being added, and the bending moments consequently go on diminishing.

As a check on the accuracy of the calculations, the bending moment at the last point N ought to be = 0.

EXAMPLE of the use of Rules E, F, and G:—Suppose a beam of 50 feet span to be supported at the ends, and loaded at four

intermediate points, so that the point N is the point 5; and let the intervals and loads be as follows:—

Points,.....	0	1	2	3	4	5
Intervals, feet,.....		10	5	5	5	25
Loads, tons,.....			5	3	3	1

Then the calculations are as follows, by Rule E:—

Loads.	Leverages about O.	Moments about O.
Tons.	Feet.	Foot-tons.
5	10	50
3	15	45
3	20	60
1	25	25

12 Resultant Load,) 180 Resultant Moment.

Distance of Resultant from O, 15 feet.

Supporting Pressure at O, $\frac{12 \times 35}{50} = 8.4$ tons.

“ “ at 5, $\frac{12 \times 15}{50} = 3.6$ tons.

By Rule F—

	Tons.
Racking force on Division 0 1,.....	+ 8.4
Load at 1,.....	— 5.0
Racking Force on Division 1 2,.....	+ 3.4
Load at 2,.....	— 3.0
Racking Force on Division 2 3,.....	+ 0.4
Load at 3,.....	— 3.0 Racking action reversed
Racking Force on Division 3 4,.....	— 2.6
Load at 4,.....	— 1.0
Racking Force on Division 4 5,.....	— 3.6

(Agreeing with the supporting pressure at 5.)

By Rule G—

Points.	Racking Forces.	Lengths of Divisions.	Products.	Bending Moments.
	Tons.	Feet.	Foot-tons.	Foot-tons.
0				0
1	+ 8.4	10	+ 84	
2	+ 3.4	5	+ 17	84
3	+ 0.4	5	+ 2	101
4	— 2.6	5	— 13	103 Grentest.
5	— 3.6	25	— 90	90
.....				0

CASE III.—Load Distributed.—Support at two points.—When the load is distributed continuously over the span of a beam, and the supporting pressures concentrated, or nearly so, at the two ends, the following is the process:—

In Fig. 7, let O and N be the points of support, and ON the

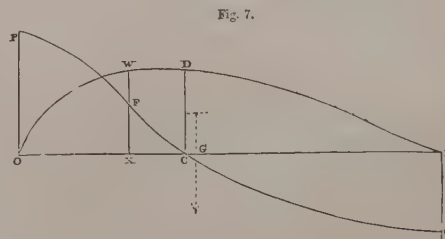


Fig. 7.

axis of the beam. The manner of distribution of the load being known, construct a curve, O W D N, such that the height of the ordinate, X W, at any point, X, of the span, shall represent the intensity of the load at that point in units of weight per unit of

* The following is the algebraical expression of the Rules E, F, and G:—

Let the distances of the points 1, 2, 3, &c., from O, be denoted by—

$x_1, x_2, x_3, \&c.$;

and the intervals between the points O, 1, 2, 3, &c., by—

$\Delta x_1 = x_1; \Delta x_2 = x_2 - x_1; \Delta x_3 = x_3 - x_2; \&c.$

Also let the forces applied at the points O, 1, 2, 3, &c., be denoted by—

$P_0, P_1, P_2, P_3, \&c.$

Those forces must necessarily fulfil the two following conditions of equilibrium—

$\Sigma P = 0; \Sigma P x = 0.$

In a beam loaded at 1, 2, 3, &c., and supported at O and N, let upward forces be positive and downward forces negative; also let

$P_1 = -W_1; P_2 = -W_2; P_3 = -W_3; \&c.$

Then P_0 and P_N are found as follows, by Rule E: make

$\frac{\Sigma W x}{\Sigma W} = x_0$; then $P_0 = (\Sigma W) \frac{x_N - x_0}{x_N}$; $P_N = (\Sigma W) \frac{x_0}{x_N}.$

Let the racking forces on the divisions $\Delta x_1, \Delta x_2, \Delta x_3, \&c.$, be denoted by $F_1, F_2, F_3, \&c.$, and the bending moments at 1, 2, 3, &c., by $M_1, M_2, M_3, \&c.$ Then by Rules F and G—

$F = \Sigma P$; and $M = \Sigma F \Delta x.$

In the case in which $P_1 = -W_1, \&c.$, these formulæ are equivalent to the following:—

$F_1 = P_0; F_2 = F_1 - W_1; F_3 = F_2 - W_2; \&c.$

$M_1 = F_1 \Delta x_1; M_2 = M_1 + F_2 \Delta x_2; M_3 = M_2 + F_3 \Delta x_3; \&c.$

The maximum value of M occurs at the point where F reverses its sign (or point of reversed racking).

The checks upon the accuracy of the calculations are expressed as follows:—

$P_N = -P_0; M_N = 0.$

span (say in pounds per foot of span). The entire area of that curve, and the area of any part of it bounded by ordinates, may be found by Simpson's Rule. The entire area will represent the whole load on the beam; and the area of any part, such as O X W, bounded by an ordinate, as X W, will represent the load upon the corresponding part, O X, of the span of the beam. Then proceed as follows:—

RULE H.—*To find the supporting pressures.*—By the Rules of Division First, Article 37, page 17, find the position longitudinally of the centre of the area O W D N; this will give the point G, where the resultant of the load cuts the axis of the beam. Then compute the supporting forces, as in Rules A and E.

RULE K.—*To compute and represent the racking forces.*—At the points O and N, draw ordinates, O P and N Q, in opposite directions, and of lengths representing the magnitudes of the two supporting forces. Those ordinates will represent the racking forces at O and N respectively. For the racking force at any intermediate point, such as X, measure by Simpson's Rule the area, O X W, which represents the load between O and X; subtract that load from the supporting force at O; the remainder will be the racking force required, which may be represented by an ordinate, X F; and when a convenient number of such ordinates have thus been found by calculation, a line, P F Q, may be drawn through their ends, to facilitate the finding of the racking force at points intermediate between the calculated ordinates.

The point C where that line cuts the axis, is the *point of reversed racking*, where the racking force becomes = 0 in the act of changing from positive to negative. The ordinate CD at that point divides the load O W D N into two parts, each of which is equal to the supporting force at the adjoining point of support; whence it follows, that the racking force at X is equal to the load between X and C, and is represented by the area W X C D.

RULE L.—*To find the Bending Moments.* The greatest bending moment is exerted at the point C, and is represented by either of the equal areas, O P C = C N Q, which may be measured by Simpson's Rule. For the bending moment at any point X between O and C, take the area O P F X; should the point be between C and N, take the area extending from the ordinate representing the racking force at the point in question to N Q.*

CASE IV.—*Load and Support both distributed.*—This is the condition of a ship supported by the pressure of the water. In

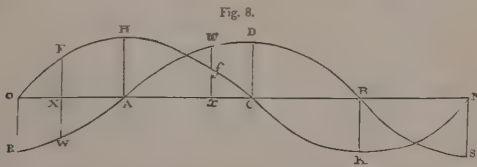


Fig. 8, let ON represent the axis of a beam; and let the ordinates of the curve, RADBS, represent the direction and intensity of pressures applied to that beam, the intensity being expressed in

* In algebraical symbols, let w denote the load per unit of span, F the racking force, and M the bending moment, at a point whose distance from O is x ; and let s denote the whole span of the beam. Then the position of the resultant is given by the equation—

$$x_c = \frac{\int_0^s x w dx}{\int_0^s w dx};$$

the supporting forces by the equations—

$$P_0 = \frac{x_s - x_c}{s} \cdot \int_0^s w dx; \quad P_s = \frac{x_c}{s} \cdot \int_0^s w dx;$$

and the racking force and bending moment at X are—

$$F = P_0 - \int_0^x w dx; \quad M = \int_0^x F dx.$$

units of force per unit of span (as lbs. per lineal foot). For example, the ordinate X W represents downward pressure, and the ordinate $x w$ upward pressure.

The figure and dimensions of the *curve of loads*, RADBS, are subject to two conditions. First, the resultant force must be nothing, and therefore the total area above the axis (A D B) must be equal to the total area below the axis (O R A + B N S); and secondly, the resultant moment must be nothing, and therefore the moments relatively to a transverse axis traversing O of the total areas above and below the longitudinal axis must be equal to each other. These conditions are fulfilled in a vessel floating steadily.

RULE M.—*To find the Racking Force at any point.*—Measure by Simpson's Rule the area of the curve of loads from O to the ordinate at the point in question; observing that areas above and below the longitudinal axis, O N, are to be regarded as of opposite signs.

For example, the racking force at X is represented by the area R O X W; but the racking force at x is represented by—

$$\text{Area R O A} - \text{area A } x w.$$

The racking forces may be represented by the ordinates of a curve, O H C K N.

At the points A and B, where the curve of loads crosses the axis O N, the racking force attains its maximum values, represented by the ordinates A H and B K, or by the areas R O A and S N B.

The point C, where the curve of racking forces crosses the axis O N, is the *point of reversed racking*, and the ordinate CD of the curve of loads at that point divides that curve in such a manner, that—

$$\begin{aligned} \text{The area A D C} &= \text{the area R O A, and} \\ \text{the area B D C} &= \text{the area S N B.} \end{aligned}$$

RULE N.—*To find the Bending Moments.*—The greatest bending moment is exerted at the point of reversed racking, C, and is represented by either of the two equal areas of the curve of racking forces, O H C = C K N. At any other point, measure the area contained between the ordinate of the curve of racking forces at that point, and that end of the beam between which and the point, C, the given point is situated.

For example, the bending moments at X and x are respectively represented by the areas O X F and O x f.†

CASE V.—*Comparison of Similarly-loaded Beams.*—**RULE P.**—When two or more beams are acted upon by similarly-distributed loads and supporting forces, the racking forces at corresponding points in the different beams are proportional to the total loads, and the bending moments at corresponding points are proportional to the products of the loads and lengths.‡

† In algebraical symbols, let p denote the pressure exerted transversely on each unit of span of the beams at a distance x from the point O, the direction of that pressure being indicated by its algebraical sign; F the racking force; and M the bending moment, at the same point.

Then the distribution of the pressure, p , is subject to the two following conditions, in which the integration extends to the whole span—

$$\int_0^s p dx = 0; \quad \int_0^s x p dx = 0;$$

The racking force at the distance x from O is—

$$F = \int_0^x p dx;$$

and the bending moment at the same point,

$$M = \int_0^x F dx.$$

The points of maximum racking force are those at which $p = 0$; and the point of reversed racking, where the bending moment is greatest, is found by putting $F = 0$, and deducing from that equation a value of x intermediate between 0 and the whole span.

‡ The following Table (from "A Manual of Civil Engineering," by the Editor of this Treatise) gives formulae for the racking force and bending moment in some of the more common cases of loaded beams. In each example in the Table, l denotes, for a beam fixed at one end, the length measured from the outer point of support to the farthest projecting loaded point, and for a beam supported at both ends, the length or span between the points of support; W denotes the total load; F and M denote the racking force and bending moment at any cross-section situated at the distance x' from the origin (which is the point where $M = 0$); F_1 denotes the greatest racking force; x'_1 the position of the section where it occurs; M_0 the

CASE VI.—*Combined Loads.*—RULE Q.—When two or more sets of forces whose effects are separately known are applied to a beam at once, the racking force at each point is the resultant of the racking forces, and the bending moment at each point is the resultant of the bending moments, which each set of forces would have produced separately.

CASE VII.—*Variable Load.*—When the forces applied to a beam are liable to variations in their amount and distribution, that amount and distribution must be taken which produce the most severe stress. Sometimes the distribution which produces the most severe racking action is not the same with that which produces the most severe bending action. The most important instance of this in practice is that to which the following rule applies:—

RULE Q.—When a beam is supported at its two ends, the greatest bending moment at each cross-section occurs when the beam is loaded over its whole span; and the greatest racking force at a given cross-section occurs when the longer of the two divisions

greatest bending moment; x'_0 the position of the cross-section where it occurs; k, m , two factors expressing respectively the ratio of the greatest racking force to the whole load, and of the greatest bending moment to the product of the load and length; that is to say,

$$F_1 = k W; M_0 = m W l \dots \dots \dots (1.)$$

Upward forces and moments are considered as positive, downward as negative.

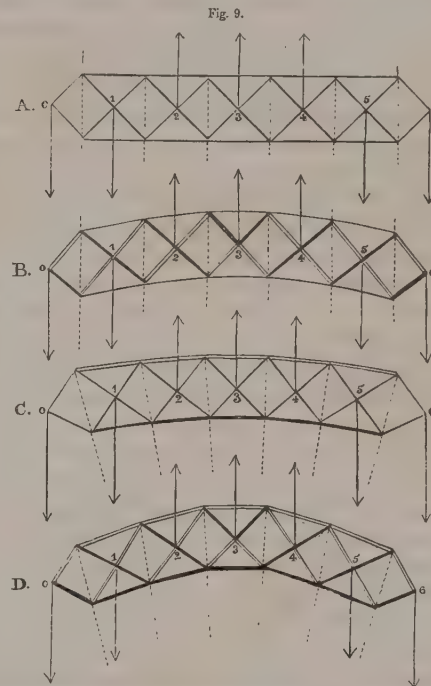
CASES.	F	x'_1	k	M	x'_0	m
A. BEAMS FIXED AT ONE END.						
I. Loaded at extreme end with W	$-W$	anywhere	-1	$-W x'$	l	-1
II. Uniform load of intensity $w = W \div l$	$-w x'$	l	-1	$-\frac{w x'^2}{2}$	l	$-\frac{1}{2}$
III. Uniform load of intensity w , and additional load W at extreme end,...	$-W' - w x'$	l	-1	$-W' x' - \frac{w x'^2}{2}$	l	$-\frac{W' + \frac{w l}{2}}{W' + w l}$
B. BEAMS SUPPORTED AT BOTH ENDS.						
IV. Single load W , in the middle;						
half of beam next origin,.....	$\frac{W}{2}$	0	$\frac{1}{2}$	$\frac{W x'}{2}$	$l/2$	$\frac{1}{4}$
further half,.....	$-\frac{W}{2}$	l	$-\frac{1}{2}$	$\frac{W (l - x')}{2}$	$l/2$	$\frac{1}{4}$
V. Single load W , applied at x' ;						
between x'' and origin;.....	$\frac{l - x''}{l} W$	anywhere	$\frac{l - x''}{l}$	$\frac{x' (l - x'')}{l} W$	x''	$\frac{x' (l - x'')}{l^2}$
beyond x'' ;.....	$-\frac{x''}{l} W$	anywhere	$-\frac{x''}{l}$	$\frac{(l - x') x''}{l} W$	x''	$\frac{(l - x') x''}{l^2}$
VI. Uniform load of intensity $w = W \div l$	$w (\frac{l}{2} - x')$	0 and l	$\pm \frac{1}{2}$	$\frac{w x' (l - x')}{2}$	$l/2$	$\frac{1}{8}$
VII. Partial load of uniform intensity $w = W \div l$ from 0 to x'' ; remainder unloaded;						
between x'' and origin;.....	$w (\frac{x''^2}{2l} - \frac{x''}{2})$	0	$1 - \frac{x''}{2l}$	$\frac{w}{2} \left\{ \frac{(x'' - x')^2}{2l} - \frac{x''^2}{2} \right\}$	x''	$-\frac{x''^2}{2l} \left(1 - \frac{x''}{2l} \right)$
beyond x'' ;.....	$-\frac{w x''^2}{2l}$			$\frac{w x''^2}{2l} (l - x')$	x''	$\frac{x''^2}{2l} (1 - \frac{x''}{2l})$

In the following example a beam supported at both ends is supposed to be loaded at a series of detached points, which divide the length of the beam into N equal divisions, so that the length of one of those divisions is $l \div N$. The origin of co-ordinates being at a point of support, the plane of section in each example is supposed to be immediately beyond the n^{th} division from that point.

CASE.	F	x'_1	k	M	x'_0	m
VIII. Each intermediate point loaded with w ; total load $(N - 1) \cdot w$	$(\frac{N-1}{2} - n) \cdot w$	0 or $N - 1$	$\pm \frac{1}{2}$	$\frac{w}{2} \left(\frac{N-n}{2} - n \right) l$	$l/2$	$\begin{cases} (N \text{ even}) \frac{1}{8} \\ (N \text{ odd}) \frac{N^2 - 1}{8 N^2} \end{cases}$

into which that cross-section divides the span is loaded, and the shorter unloaded.*

45. *Resistance of a Skeleton Beam.*—In order to make clear the manner in which beams resist the racking and bending actions of the forces applied to them, the first case explained will be that of a skeleton beam, composed of a framework of slender bars of the simplest possible construction: that is to say, of a pair of parallel longitudinal pieces, which may be called the *stringers*, connected together by a series of diagonal pieces, or *braces*. Diagram A, of Fig. 9, represents such a skeleton beam, with the stringers horizontal; the braces are all equally



inclined to the horizon at angles of 45° ; and the forces with which the beam is loaded are supposed to be applied to the points of intersection of the braces, 0, 1, 2, 3, 4, 5, 6: that being the mode of application whose effects on the beam are most simple.

The forces are supposed to be so distributed (as shown by the arrows), that they tend to make the originally straight beam become convex upwards.

In a skeleton beam such as is here represented, the whole of the racking force on each division of the span is resisted by the braces, and the whole of the bending moment at each loaded point is resisted by the stringers; and the effects of those two kinds of action may be considered separately, as follows:—

Diagram B shows, on an exaggerated scale, the *separate effect of the racking action*, which is to lengthen one set of braces, and shorten another; thus racking or rendering oblique-angled the originally square parallelograms of which those braces are the

* Let l denote the span of a beam supported at the two ends, and x the length of a division which is uniformly loaded with w' lbs. per unit of span, the division $l - x$ being unloaded. Then the shearing force F' at the boundary between the loaded and unloaded divisions is given by the following formula—

$$F' = \frac{w' x^2}{2l}.$$

If the whole span be uniformly loaded at the rate w , and the division x at the rate w' , the racking force at the same boundary is expressed as follows—

$$F + F' = w \left(x - \frac{l}{2} \right) + \frac{w' x^2}{2l}.$$

diagonals. The stretched braces, or *diagonal ties*, are shown by double lines: the compressed braces, or *diagonal struts*, are shown by thick black lines.

The determination of the stress produced on each brace by the racking action of the forces depends on the following principles:—

In any given division of the span (such, for example, as 1 2) the horizontal components of the stresses along the braces are equal, and they oppose and balance each other: the vertical components of those stresses are also equal, and they concur with each other to oppose the racking force, each of them balancing one half of it. Hence follows:—

RULE I.—To find the stress along each diagonal brace of a given division of the span:—*Multiply one-half of the racking force by the ratio in which the length of a brace is greater than the vertical depth between its ends: that being the ratio in which the total stress along a brace is greater than the vertical component of that stress.* (In the present example, where the inclination of each brace is 45° , that ratio is $\sqrt{2}$.)

Each brace is to be made of dimensions sufficient to bear safely the stress thus determined, by the rules of Section III. or of Section IV., according as it is a tie or a strut.

In *Lattice Beams*, where more than one pair of braces are intersected by each cross-section, it is sufficiently accurate for practical purposes to conceive the racking force as being equally shared amongst the braces so intersected. In the *Warren* or *zig-zag* girder, the whole racking force at each cross-section is borne by one brace.

Diagram C shows (upon an exaggerated scale also) the *separate effect of the bending moment*, which is to stretch the convex stringer, and compress the concave stringer; the originally square divisions of the beam becoming wedge-shaped. As before, the stretched bars are marked by double lines, and the compressed bars by thick black lines.

At each cross-section of the beam made by a transverse plane traversing one of the loaded points, the tension along the convex stringer and the thrust along the concave stringer are equal and opposite, and form a couple of stresses, whose arm or leverage is the *depth of the beam*, measured from centre to centre of the stringers. The moment of that couple is the *moment of resistance* of the beam at the cross-section in question; and it is equal and opposite to the bending moment, which it balances. Hence follows:—

RULE II.—To find the amount of the stress along each of the stringers at a given cross-section, *divide the bending moment at the given cross-section by the depth of the beam from centre to centre of the stringers.*

In making use of this rule, care must be taken that the depth is expressed in the same units of measure with the span and its divisions, employed in computing the bending moment.

Each piece of each of the stringers is to be made sufficient to bear safely the stress thus determined, by the rules of Section III. or of Section IV., according as the piece in question is a tie or a strut.

Diagram D shows (still on an exaggerated scale) the *combined effects of the racking forces and bending moments*; and, as before, the stretched bars are marked by double lines, and the compressed bars by thick black lines.

I-Shaped Beam.—There is one form of continuous girder whose condition very often nearly approximates to that of a skeleton

beam; and that is the I-shaped beam whose cross-section is represented by Fig. 10, and which consists of a pair of parallel stringers or flanges, A and B, connected together by a thin web C. Strictly speaking, the web assists the stringers in resisting the bending action; and when the sectional area of the web is considerable compared with that of the stringers, that part of the resistance should not be neglected; but it may be neglected when the sectional area of the web is comparatively small; and then *the whole bending moment may be regarded as resisted by the flanges or stringers*, the stress along which is to be computed by Rule II.,^{*} and *the whole racking force may be regarded as resisted by the web.* The racking stress at each cross-section of the web may also in this case be regarded as uniformly distributed; whence follows:—

RULE III.—To find the intensity of the racking stress at a given cross-section of the web of an I-shaped girder, *divide the racking force by the depth and by the thickness of the web.*

It is to be observed that this is the intensity of the stress tending to detach the web from the stringers at its upper and lower edges; and if there are fastenings, such as rivets, at those edges, they must be of dimensions sufficient to bear that stress with safety.

The sectional area of the web is usually considerably greater than is necessary to resist the tendency of the racking force to tear it in a diagonal direction; but in order to enable a deep and thin web to resist the tendency of the racking force to crush it by diagonal buckling, it is often requisite to stiffen it by means of ribs. The most efficient position for those ribs would be diagonal, like that of the strut-braces in a skeleton beam, and their state of stress would be the same; but for reasons of practical convenience, they are usually placed upright, and then their effect may be estimated as follows:—

In applying Rule I. of Article 37 to the calculation of the resistance of the web to buckling under the racking force, consider the *length of the piece* as being represented by the *distance between the stiffening ribs measured on a slope of 45°* , and the *least diameter of the piece* as being represented by the *thickness of the web.*

RULE IV.—*The moments of resistance of different skeleton beams or thin-webbed I-shaped beams of the same material, are to each other as the sectional areas of the stringers and as the depths of the beams.*

46. Resistance of a Continuous Beam to a Bending Moment.—In Fig. 11, let *ab* represent a cross-section of a continuous beam, at which cross-section a given racking force and bending moment are to be resisted; and let *ABBA* represent a small portion

Fig. 10.

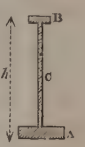
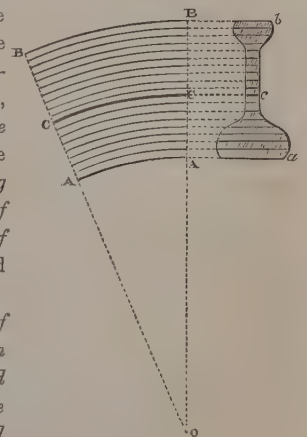


Fig. 11.



* As to the application of Rule I. of Article 37 to the calculation of the strength of the compressed flange of an I-shaped wrought iron beam, the following consequences have been deduced from Mr. Fairbairn's experiments: divide the square of the span of the beam by the square of the breadth of the compressed flange, and by 5000; to the quotient add 1; divide 86,000 by the sum; the result will give the ultimate strength of the flange in lbs. per square inch of section. (See Rankine on Civil Engineering, pp. 527, 529.)

of a longitudinal section of the same beam near the given cross-section, divided into originally straight layers which have become curved through the action of the load, *O* being the centre of curvature of those layers at the given cross-section.

The layers at and near the convex side of the beam, *BB*, are longitudinally stretched. The layers at and near the concave side of the beam, *AA*, are longitudinally compressed; and a certain intermediate layer, *CC*, called the *neutral layer*, is neither stretched nor compressed. Thus the layers between *CC* and *BB* perform the duty of the extended stringer of a skeleton beam, and those between *BB* and *AA* the duty of the compressed stringer, in resisting the bending moment; but with this modification—that the outermost layers only, *BB* and *AA*, are capable of exerting their full strength; for the inner layers, besides acting with less leverage, are less strained than the outermost layers, in proportion as they are less distant from the neutral layer *CC*.

The action of the racking force on the cross-section is distributed in such a manner that the intensity of the racking stress which resists it is *greatest at the neutral layer CC*, and diminishes to nothing at the surfaces *AA* and *BB*.

As the straining effects of the racking force are more complex than those of the bending moment, and as they may in many practical cases be disregarded on account of their comparative smallness, their further consideration is deferred to Article 47.

The resistance of a given cross-section of a beam to a bending moment is determined by the aid of the following principles:—

The amount of the direct stress exerted along a given layer is proportional to the sectional area of that layer, and to its distance from the neutral layer—in other words, to its *geometrical moment* relatively to the neutral layer (see Division I., Article 34, page 16); and the resultant stress exerted in the form of tension by the layers from *C* to *B*, and thrust by the layers from *B* to *A*, is nothing; therefore the neutral layer is so situated that the geometrical moment of the cross-section *ab* relatively to that layer is nothing; that is to say, the layer traverses the centre of that cross-section. The line in which the neutral layer cuts a cross-section is called the *neutral axis* of that section, and is found as follows:—

RULE I.—*Find, by the rules of Division I., Article 37, page 17, the centre of the cross-section; a line traversing that centre transversely will be the neutral axis.*

In a cross-section that is symmetrical above and below, the neutral axis is obviously at the middle of the depth. In a skeleton beam, the neutral axis is an imaginary line, dividing the depth in the inverse ratio of the sectional areas of the two stringers.

The statical moment of the direct stress exerted by a given layer is the product of the area of the layer, the intensity of the stress, and the distance of the layer from the neutral axis. The intensity of the stress at the given layer is equal to the intensity of the stress at the most severely strained layer (*b*, in Fig. 11) multiplied by the ratio in which the distance of the given layer from the neutral axis is less than the distance of the most severely strained layer from the neutral axis. Therefore the statical moment of the stress at the given layer has the following value: (1) *the intensity of the stress at the most severely strained layer, multiplied by* (2) *the sectional area of the layer under consideration, multiplied by* (3) *the square of the distance of that layer from the neutral axis, and* (4) *divided by the distance of the most severely strained layer from the neutral axis.* But the product of the second and third of those

factors is what has been already explained (in Division I., Article 43, page 20) to be the *geometrical moment of inertia* of the sectional area of the layer; and the sum of similar products for all the layers is the *geometrical moment of inertia of the whole cross-section*, about the neutral axis; whence follows—

RULE II.—*To find the moment of resistance of a given cross-section of a beam: divide the moment of inertia of the cross-section about the neutral axis, by the distance of the most severely strained layer from that axis, and multiply the quotient by the stress on that layer.* The *ultimate moment of resistance* is the value of the above quantity, when the greatest intensity of stress is that which is just sufficient to cause breaking to commence. The value of that intensity is called the *Modulus of Rupture*.

A beam, in the act of breaking, gives way by tearing at the convex side, *BB*, or by crushing at the concave side, *AA*, according as the stress at the one side or the other soonest reaches the breaking intensity; and that depends partly on the nature of the material, and partly on the form of cross-section; so that the *modulus of rupture* represents either the ultimate tenacity of the most severely stretched layer, or the crushing stress of the most severely compressed layer, according as the beam gives way by tearing or by crushing, and has different values for the same material under those different circumstances. When the section is symmetrical above and below, so that the neutral axis is at the middle of the depth, the beam gives way by tension or by compression, according as the tenacity or the resistance to crushing is the less; for example, a cast-iron beam gives way by tearing, a wrought iron beam by crushing.

In a skeleton beam, the modulus of rupture is simply equal to the tenacity or the crushing stress of a separate bar of the material, as the case may be; and such is nearly the case in thin-webbed I-shaped beams also. But in continuous beams of more full forms of section (such as rectangular beams) the resistance of the particles of the outermost convex and concave layers is modified, and in general increased, through their connection with the inner layers, which are less severely strained; so that the modulus of rupture in such beams is often different both from the direct tenacity and from the resistance to crushing, and can be ascertained by special experiments only. Numerous examples of this are afforded by the Tables of the Strength of Materials.

In calculating the moments of resistance of beams of the more ordinary and simple forms of cross-section, time and labour are saved by making use of the principles, that *the moments of inertia of similar cross-sections are proportional to their areas and the squares of their depths*, or otherwise, *to their breadths and the cubes of their depths*; and that, *in similar cross-sections, the distances of the most severely strained layer from the neutral axis bear the same ratio to the depths*; from the combination of which principles it follows, that *the moments of resistance of beams of the same material and of similar cross-sections, are proportional to their breadths and to the squares of their depths*, or in other words, *to their sectional areas multiplied by their depths*.

These principles, being reduced to practical rules, take the following form:—

To find the ultimate moment of resistance of a given cross-section of a beam—

RULE III.—*Multiply the square of the depth by the breadth, by a factor depending on the figure of the cross-section, and by the modulus of rupture; or otherwise—*

RULE IV.—Multiply the sectional area by the depth, by a factor depending on the figure of the cross-section, and by the modulus of rupture.

The following Table gives the factors suitable for a variety of ordinary forms of cross-section. Algebraical symbols are necessarily given for some of the factors, because they cannot conveniently be expressed otherwise:—

Form of Cross-section.	Factor (n) to multiply Breadth × Depth. ²	Factor (g) to multiply Area × Depth.
I. Rectangle, solid,.....	$\frac{1}{6}$	$\frac{1}{6}$
II. Ellipse and Circle, solid,	$\frac{\pi}{32} = .0982$	$\frac{1}{8}$
III. Rectangle, hollow; breadths; outside b ; inside b' ; depths; outside h ; inside h' ; also I-shaped section with equal stringers, if depth of web = h' , thickness $b - b'$,.....	$\frac{b h^3 - b' h'^3}{6 h^3}$	$\frac{(b h^3 - b' h'^3)}{6 h^3 (b h - b' h')}$
IV. Square, hollow; dimensions, outside, $h \times h$; do. inside, $h' \times h'$,.....	$\frac{h^4 - h'^4}{6 h^3}$	$\frac{h^3 + h'^3}{6 h^2}$
V. Square, hollow; very thin; dimensions $h \times h$; thickness, t ,.....	$\frac{4t}{3h}$	$\frac{1}{3}$
VI. Ellipse, hollow; symbols the same as for hollow rectangle,.....	$.0982 \cdot \frac{b h^3 - b' h'^3}{b h^3}$	$\frac{(b h^3 - b' h'^3)}{8 h^3 (b h - b' h')}$
VII. Circle, hollow; diameter, outside, h ; do. inside, h' ,.....	$.0982 \cdot \frac{h^4 - h'^4}{h^3}$	$\frac{h^3 + h'^3}{8 h^2}$
VIII. Circle, hollow; very thin; diameter, h ; thickness, t ,.....	$\frac{7854 t}{h}$	$\frac{1}{4}$
IX. Isosceles Triangle,.....	$\frac{1}{12}$	$\frac{1}{12}$
X. I-shaped section; area of compressed stringer, A; area of stretched stringer, B; area of web, C. (Approximate formulae.) If the beam gives way by tearing the stringer B,.....	—	$\frac{C(C + 4A + 4B) + 12AB}{6(C + 2A)(A + B + C)}$
XI. The same, If the beam gives way by crushing the stringer A,.....	—	$\frac{C(C + 4A + 4B) + 12AB}{6(C + 2B)(A + B + C)}$
XII. Barlow Rail,*.....	—	$\frac{2}{3}$
XIII. Buckled and Corrugated Plates,.....	—	$\frac{1}{15}$

For T-shaped beams, the formulæ of Cases X. and XI. are to be used, making either $A = 0$ or $B = 0$, according as it is the stretched or the compressed edge of the web that is provided with a flange.

In designing a beam, the use to be made of the factors given in the preceding Table is as follows:—

Having determined the greatest bending moments due to the dead load and working live load respectively, multiply these by suitable factors of safety (say, in the case of iron beams, 3 for the dead load, and 6 for the working live load), and add together the products; their sum will be the required ultimate moment of resistance at the most severely strained cross-section.

* A "Barlow Rail" is a rolled iron bar, the cross-section of which consists of a pair of quadrantal wings spreading concavely outwards from a flat head. In the Table, the sectional area of the head is supposed to be 0.273 of the joint area of the two wings, or .215 of that of the whole bar; and when such is the case, the sectional area of the bar is four times the product of the thickness and radius of the wings, the radius being measured to the middle of the thickness.

The depth is usually fixed so as to fulfil conditions of stiffness to be explained further on; then employ one or other of the following rules:—

RULE V.—To find the breadth: divide the ultimate moment of resistance by the square of the depth, the modulus of rupture, and the factor (n) suited to the intended form of section.

RULE VI.—To find the sectional area: divide the ultimate moment of resistance by the depth, the modulus of rupture, and the factor (q) suited to the intended form of section.

EXAMPLE, suppose—

Bending moment due to dead load,.....	33 Foot-tons.
" " " " live load,.....	70 " "
83 × 3 (factor of safety), =	99 " "
70 × 6 (factor of safety), =	420 " "
Required ultimate moment of resistance,.....	519 " "
	× 26,880
The same, reduced to inch-lbs.,.....	13,950,720

Required (1) the breadth of a solid rectangular beam of forged iron to bear the above load, the depth being 12 inches, and the modulus of rupture 40,000 lbs. on the square inch.

	Ult. mom. of res.
Divide by the square of the depth,.....	144) 13,950,720
Divide by the modulus of rupture,.....	40,000) 96,880
	2.422
Divide by the factor, $n = \frac{1}{6}$; that is to say, multiply by	6
Breadth required,.....	14.532 inches.

Required (2) the sectional area of an I-shaped wrought-iron beam to bear the same load, the depth being 20 inches, the modulus of rupture 30,000 lbs. on the square inch of thrust along the compressed stringer, and the areas of the stringers and web proportioned as follows:—

Compressed stringer A : stretched stringer B : web C
: : 7 : 3 : 6.

Computation of the factor, q, by Case XI. of the Table:—

$$q = \frac{6(6 + 4 \times 7 + 4 \times 3) + 12 \times 7 \times 3}{6(6 + 2 \times 3)(7 + 3 + 6)} = \frac{408}{1152} = \frac{17}{48}$$

	Ult. mom. of res.
Divide by the depth,.....	20) 13,950,720
Divide by the modulus of rupture,.....	30,000) 697,536
	23.2512
Multiply by denominator of factor, q,.....	48
Divide by numerator of factor, q,.....	17) 1116.0576
Total sectional area required, A + B + C =	65.65 sq. inches.

$$\begin{aligned} \text{Sectional area of stringer A} &= \frac{7}{18} (A + B + C) = 28.72 \\ \text{" " stringer B} &= \frac{3}{18} (A + B + C) = 12.31 \\ \text{" " web C} &= \frac{6}{18} (A + B + C) = 24.62 \end{aligned}$$

In cases in which there is difficulty in distinguishing between the dead and the live load, it is often convenient to modify Rules II., III., IV., V., and VI., as follows:—

RULE VII.—Instead of the ultimate moment of resistance, use the greatest working moment; and instead of the modulus of rupture, use a working modulus of resistance (equal to the modulus of rupture divided by a suitable factor of safety); then proceed as in Rules II., III., IV., V., or VI.

The factor of safety used in the practical application of this rule is sometimes equal to, and sometimes rather less than, that applicable to a wholly live load; as is shown by the following values

of the working modulus, commonly employed in engineering structures:—

	Working Modulus; lb. on the sq. inch.	Factor of Safety.
Wrought iron girders; parts under tension,.....	11,200 (= 5 tons) ...	about 5
" " " parts under compression,...	8,960 (= 4 tons) ...	about 4
Structures in strong timber, as oak, elm, pine, &c.,	1,000	... about 10

In most questions respecting the strength of ships, it is advisable (as will appear more fully in a future Chapter) to consider *the whole load as live load*; and then a working modulus may be used, found by dividing the modulus of rupture by a factor of safety suitable to a live load: for example;—

	Modulus of Rupture; lb. on the sq. inch.	Factor of Safety.	Working Modulus.
Wrought iron under tension,.....	48,000 to 60,000	6	8,000 to 10,000
" " under compression, 24,000 to 36,000	6	4,000 to 6,000
Timber (good average),.....	10,000	10	1,000*

47. *Distribution of Racking Stress in a Beam.*—The intensity of the *Racking Stress* at any given layer of a given cross-section of a beam is found by the following process; for the demonstration of which see Rankine "On Applied Mechanics," page 339.

Find the *geometrical moment* (Div. I., Art. 37, page 17), relatively to the neutral axis, of that part of the cross-section which lies between the given layer and that outside edge of the cross-section which is at the same side of the neutral axis with the given layer.

Divide that moment by the breadth of the given layer, and by the geometrical moment of inertia of the whole cross-section; multiply the quotient by the amount of the racking force on the given cross-section; the product will be the required intensity of the racking stress.

The *greatest* intensity of the racking stress at a given cross-section occurs at the neutral axis. The *mean* intensity of the racking stress at a given cross-section is found by dividing the

* The algebraical expression of the principles of the resistance of a beam to the bending moment exerted at a given cross-section is as follows:—

Conceive the cross-section, as in Fig. 11, to be divided into an indefinite number of thin layers, each of the depth dy ; let z be the breadth of any one of those layers; then—

$$\text{Whole area of section} = \int z dy.$$

Assume any convenient transverse axis; for example, the compressed edge a of the cross-section; let y' be the distance of any layer, $z dy$, from that edge, and y_a the distance of the neutral axis from the same edge; then the position of the neutral axis is given by the formula—

$$y_a = \frac{\int y' z dy}{\int z dy} \quad \text{..... (Rule I.)}$$

Now let the axis of co-ordinates be shifted to the neutral axis, and let y be the distance of any layer, $z dy$, from that axis, so that—

$$y = y' - y_a.$$

Find, by the Rules of Division I., Article 48, pages 20, 21, the geometrical moment of inertia of the cross-section about its neutral axis, and let it be denoted by I ; then—

$$I = \int y^2 z dy.$$

Let y_a be the distance of the most severely compressed layer from the neutral axis;

y_b that of the most severely stretched layer;

f_a the modulus of resistance at the compressed side of the beam;

f_b the modulus of resistance at the stretched side.

Then the beam will tend to give way by crushing or by tearing, according as—

$$\frac{f_a}{y_a} \text{ or } \frac{f_b}{y_b}$$

is the less.

Let f stand for f_a or f_b , as the case may be, and y , for y_a or y_b , as the case may be. Then the *Moment of Resistance* is—

$$M = \frac{fI}{y} \quad \text{..... (Rule II.)}$$

Let b denote the extreme breadth,

$h = y_a + y_b$ the total depth of the beam, and

S , its sectional area.

Then the factors n and q in the Table have the following values—

$$n = \frac{I}{y_a b h^2}; \quad q = \frac{I}{y_b h^2 S};$$

and the rules for their use take the following form—

$$M = f n b h^2 = f q h S \quad \text{..... (Rules III. and IV.)}$$

$$b = \frac{M}{f n h^2} \quad \text{..... (Rule V.)}$$

$$S = \frac{M}{f q h} \quad \text{..... (Rule VI.)}$$

According as M denotes the ultimate or the working moment of resistance, f is to be held to stand for the modulus of rupture or for the working modulus.

amount of the racking force by the area of the section. The greatest intensity exceeds the mean in a proportion depending on the figure of the cross-section, of which some examples have been stated in Article 43 of this Division.†

The following Table gives examples of the proportions in which the greatest racking stress is greater than the mean racking stress, in cross-sections of different figures. Some of these proportions are necessarily expressed in algebraical symbols, because they cannot conveniently be expressed in any other way:—

Figure of Cross-section.	Ratio $\left(\frac{\epsilon_0 S}{F}\right)$
I. Rectangle, solid,.....	$\frac{3}{8}$
II. Ellipse and Circle, solid,.....	$\frac{3}{8}$
III. Rectangle, hollow; Outside dimensions, $b \times h$; Inside dimensions, $b' \times h'$;	$\frac{3(bh - b'h')(bh^2 - b'h'^2)}{2(b - b')(b^2 - b'^2)}$
IV. Square, hollow; Outside dimensions, $h \times h$; Inside dimensions, $h' \times h'$;	$\frac{3}{2} \left(1 + \frac{h'h'}{h^2 + h'^2}\right)$
V. Square, hollow; very thin,.....	$\frac{3}{2}$
VI, VII. Ellipse and Circle, hollow; the numerical factor $\frac{3}{8}$; the factors depending on the dimensions, the same as for the hollow rectangle and square respectively.	
VIII. Circle, hollow; very thin,.....	2.
IX. I-shaped section; sectional areas of stringers or flanges, A, B; of web, C; $\frac{12(A + B + C)(2AB + AC + BC) + 3C^2}{2C(12AB + 4AC + 4BC + C^2)}$	
X. T-shaped section; area of flange or stringer, B; of web, C;.....	$\frac{12(B + C)B + 3C^2}{2C(4B + C)}$
XI. I-shaped section; the area of the web very small compared with that of the stringers,.....	$\frac{A + B + C}{C}$

The last case (XI.) is the same with that already mentioned in Article 45, Rule III., in which the whole racking force may be regarded as borne by the web, and the racking stress is uniformly distributed.

EXAMPLE of Case IX.—Let $A : B : C :: 7 : 3 : 6$; then $\frac{\phi_0 S}{F} = \frac{20232}{6336} = 3.2$ nearly = ratio in which the greatest racking stress exceeds the mean.

Direct Vertical Stress exists in one direction or another at every particle of a beam; but its intensity is in general insignificant in practice except at those points where loading or supporting forces are concentrated.

It appears, then, that except at the cross-section of greatest bending moment, where (as already shown in Article 44) there is no racking action, every particle of a beam is under the combined action of a longitudinal stress, due to the bending moment; of a vertical stress, due to the direct action of a loading or sup-

† In algebraical symbols, let—

F be the amount of the racking force;

z , the breadth, and dy , the depth of any layer of the cross-section;

y , its distance from the neutral axis;

y_1 , the distance from the neutral axis to that surface of the beam which is at the same side of the neutral axis with the given layer;

I , the moment of inertia of the whole cross-section;

ϵ , the required intensity of the shearing stress; then—

$$\epsilon = \frac{F}{Iz} \int_y^{y_1} y z dy.$$

Let ϵ_0 be the greatest value of ϵ ; then—

$$\epsilon_0 = \frac{F}{Iz_0} \int_0^{y_1} y z dy,$$

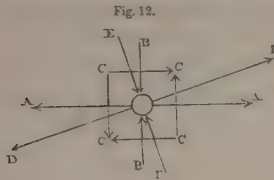
where z_0 is the breadth of the neutral layer.

Let S be the area of the cross-section; then the mean intensity of the racking stress is $\frac{F}{S}$; and the greatest intensity exceeds the mean in the ratio:—

$$\frac{\epsilon_0 S}{F} = \frac{S}{Iz_0} \int_0^{y_1} y z dy.$$

porting force; and of a racking stress, due to the racking force. The result of such a combined action is as follows:—

In Fig. 12, let \odot represent a particle, through which there are exerted a longitudinal stress, represented by the arrows, A, A, and a direct vertical stress, represented by the arrows, B, B. In the

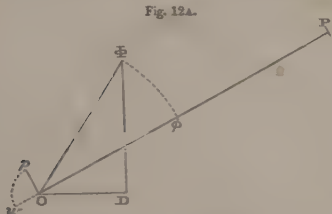


example represented by the figure, the longitudinal stress is tensile, and the vertical stress compressive; but either or both of them may be reversed. With those stresses let there now be

combined a racking stress, represented by the arrows C, C, C, C. The effect of the whole combination is the same with that of a new pair of direct stresses, D, D, E, E, at right angles to each other, but exerted in directions oblique to the original stresses, in such a manner, that according as the greatest direct stress is $\left\{ \begin{array}{l} \text{tensile} \\ \text{compressive} \end{array} \right\}$ it is made by the combination of the racking stress with it to deviate towards the $\left\{ \begin{array}{l} \text{stretched} \\ \text{compressed} \end{array} \right\}$ diagonal of a particle under the racking stress alone.

The intensity of the new direct stresses, and their angle of deviation, are found as follows:—

In Fig. 12A, draw \overline{OD} longitudinally, and of a length to represent the $\left\{ \begin{array}{l} \text{half-difference} \\ \text{half-sum} \end{array} \right\}$



of the intensities of the original stresses according as they are of $\left\{ \begin{array}{l} \text{the same kind} \\ \text{opposite kinds} \end{array} \right\}$. Perpendicular to \overline{OD} , draw $D\Phi$, to represent the intensity of the racking stress. Join $O\Phi$.

Bisect the angle $D\Phi O$ by the straight line OP . This will be the deviated direction of greatest stress.

In OP , lay off $\overline{OP} = \overline{O\Phi}$. Then from the point ϕ , lay off in opposite directions along the same line, $\phi\overline{P} = \phi\overline{\omega}$, to represent the $\left\{ \begin{array}{l} \text{half-sum} \\ \text{half-difference} \end{array} \right\}$ of the original stresses according as they

are of $\left\{ \begin{array}{l} \text{the same kind} \\ \text{opposite kinds} \end{array} \right\}$. Then \overline{OP} , and $\overline{O\omega}$ perpendicular and equal to $\overline{O\omega}$, will represent in direction and intensity the new direct stresses.*

In many cases the vertical stress may be neglected; and then $\overline{OD} = \phi\overline{P} = \phi\overline{\omega}$ will represent simply one-half of the intensity of the longitudinal stress.

* The algebraical expression of this is as follows. Let p_x, p_y denote the intensities of the original longitudinal and vertical stresses; e , the intensity of the racking stress combined with them, and p_1, p_2 the intensities of the new or resultant direct stresses, and θ , the angle ($\angle DOP$ in Fig. 12) by which they deviate from the original direct stresses. Then—

$$p_1 = \frac{p_x + p_y}{2} + \sqrt{\left\{ \left(\frac{p_x - p_y}{2} \right)^2 + e^2 \right\}};$$

$$p_2 = \frac{p_x + p_y}{2} - \sqrt{\left\{ \left(\frac{p_x - p_y}{2} \right)^2 + e^2 \right\}};$$

$$\tan. 2\theta = \frac{2e}{p_x - p_y}.$$

When the vertical stress may be neglected, those expressions become—

$$p_1 = \frac{p_x}{2} + \sqrt{\left(\frac{p_x^2}{4} + e^2 \right)};$$

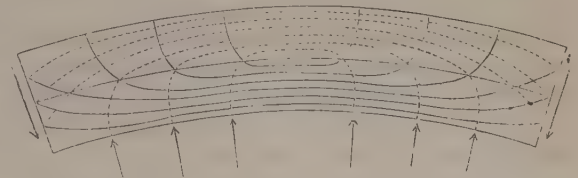
$$p_2 = \frac{p_x}{2} - \sqrt{\left(\frac{p_x^2}{4} + e^2 \right)};$$

$$\tan. 2\theta = \frac{2e}{p_x}.$$

and in such cases, p_2 is always negative, or of the contrary kind to p_x and p_1 .

If the directions OP and $O\omega$ (or *Axes of Stress*, as they are called) be determined for a great number of particles of a beam, and laid down on a drawing, a network consisting of two series of curved lines crossing each other at right angles, as in Fig. 13,

Fig. 13.



may be drawn, so that each curve shall touch the axes of stress traversing a series of particles, and so that the tangents to the pair of curves which cross each other at a given particle shall be the axes of stress of that particle.

Those curves may be called the *curves of principal stress*. At the cross-section of greatest bending moment, where there is no racking action, they are all horizontal; and they all cut the neutral layer at angles of 45° .

At every point of the outside surface of the beam where it is not subject to the action of an external racking force, the two sets of curves of principal stress are respectively parallel and normal to that surface.

In the example represented by Fig. 13, curves along which tension is exerted are dotted; those along which thrust is exerted are plain. The beam is supposed to have external racking forces applied to its ends, for which reason the curves meet the ends of the beam obliquely.†

In a beam of an I-shaped cross-section with a thin web, the curves of principal stress throughout the web coincide very nearly with straight lines inclined at 45° . At the upper and lower edges of the web they turn suddenly, and become nearly horizontal and vertical throughout the stringers.

48. *Sections of Uniform Strength* are of two kinds—cross-sections, and longitudinal sections.

A *Cross-section* of uniform strength for a beam, is one that is so proportioned that the beam has

neither more nor less tendency to break by tearing at the stretched side, than by crushing at the compressed side. There is thus no waste of material at either side of the beam. In order to design such a cross-section, the neutral axis should be nearest that side whose modulus of strength is least, and in such a position as to divide the depth of the beam into two parts, proportional to the moduli of rupture by tearing and by crushing respectively.

Fig. 14.



That is to say (in Fig. 14), let AB denote the depth of the beam, and CA and CB the two parts into which the neutral axis divides that depth; then we should have—

† The general character of the curves of principal stress in beams, and the general method of determining their figures, are demonstrated in "A Manual of Applied Mechanics," by the Editor of this Treatise, first published in 1858. A more elaborate investigation of their properties, with the results in several special cases, appeared in a paper by G. B. Airy, Esq., Astronomer-Royal, published in the Phil. Trans. for 1862.

Modulus of Rupture by crushing. } : { Modulus of Rupture by tearing. } :: $\overline{CA} : \overline{CB}$.

For example, in a certain sort of cast iron, the resistance to crushing is four times the tenacity; and consequently, in a cross-section of uniform strength—

$$\overline{CA} = \frac{4}{5} \overline{AB} ; \quad \overline{CB} = \frac{1}{5} \overline{AB}.$$

In a certain sort of wrought iron, the resistance to crushing is three-fifths of the tenacity; therefore in a cross-section of uniform strength—

$$\overline{CA} = \frac{3}{8} \overline{AB} ; \quad \overline{CB} = \frac{5}{8} \overline{AB}.$$

The forms usually employed in such beams are, the skeleton beam, the T-shaped cross-section, and the I-shaped cross-section; and the following are the rules for adjusting their proportions:—

RULE I.—In a skeleton beam, make the sectional areas of the stringers inversely proportional to their moduli of strength.

This rule may be applied also to I-shaped beams, where the web is so thin as not to assist materially in resisting the bending moment.

RULE II.—In a T-shaped beam, the stringer or flange should run along the weaker edge of the web; and its sectional area should be to that of the web, as the difference between the two moduli of strength is to twice the smaller modulus.

For example, in the cast iron before mentioned, the flange should run along the stretched side of the web; and—

$$\frac{\text{Area of flange}}{\text{Area of web}} = \frac{4-1}{2 \times 1} = \frac{3}{2}.$$

In the wrought iron before mentioned, the flange should run along the compressed edge of the web; and—

$$\frac{\text{Area of flange}}{\text{Area of web}} = \frac{5-3}{2 \times 3} = \frac{1}{3}.$$

RULE III.—In an I-shaped beam, the greater stringer or flange should run along the weaker edge of the web; and its sectional area should be the sum of two parts, one computed from the area of the smaller stringer or flange by Rule I., and the other from the area of the web by Rule II.

For example, in the cast iron before mentioned, the stretched flange should be the greater; and its sectional area should be—

$$4 \times \text{Area of Compressed flange} + \frac{3}{2} \times \text{area of web}.$$

In the wrought iron before mentioned, the compressed flange should be the greater; and its sectional area should be—

$$\frac{5}{3} \times \text{Area of Compressed flange} + \frac{1}{3} \times \text{area of web}.$$

In estimating the modulus of rupture of the compressed stringer or flange, especially in the case of wrought iron, regard should be had to its tendency to give way to the thrust along it by bending or swerving sideways; and for that purpose use is to be made of the rules of Section IV. of this Chapter, Article 38, and of the additional rule for I-shaped beams at the end of Article 45.

The following Table (extracted from "A Manual of Civil Engineering") shows the results of applying those principles to beams of such a quality of wrought iron, that its tenacity is 50,000 lbs. on the square inch, and its resistance to direct crushing 36,000 lbs. on the square inch.

In the last column, A denotes the area of the compressed flange, B that of the stretched flange, and C that of the web:—

Ratio of Span to breadth of Compressed Flange.	Modulus of Rupture by Compression; lbs. per sq. in.	Sectional Area of Compressed Flange.
		$A =$
10	35,294	$1.41 B + 0.21 C$
20 ..	33,333	$1.50 B + 0.25 C$
30	30,509	$1.64 B + 0.32 C$
40	27,273	$1.83 B + 0.41 C$
50	24,000	$2.08 B + 0.54 C$

To make a longitudinal section of uniform strength, the dimensions of the cross-section of a beam are varied at different parts of its span, so as to make the safe moment of resistance of the cross-section bear everywhere the same proportion to the bending moment exerted by the load. The object of this is to save the waste of material which takes place in a beam of uniform cross-section, from the strength of that section being too great at every point where the bending moment is less than the maximum.

At those parts of the beam where the bending moment is very small, care must be taken not to reduce the transverse dimensions below what are required to resist the racking action, and the direct vertical stress. Subject to that limitation, the required variation of the moment of resistance may be obtained as follows:—

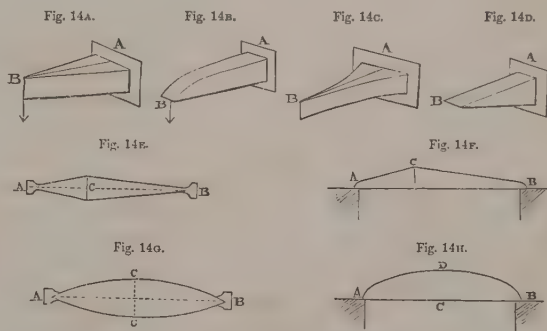
METHOD I.—The depth being uniform, make the area of each cross-section (or, in skeleton beams, of the stringers) proportional to the bending moment. (This is applicable to cast iron beams, built beams of wrought iron, and skeleton beams of all materials.)

METHOD II.—The area of cross-section being uniform (or, in skeleton beams, the area of the stringers) make the depth proportional to the bending moment. (Applicable to thin-webbed cast iron and built wrought iron beams, and to all skeleton beams.)

METHOD III.—The breadth being uniform, make the depth proportional to the square root of the bending moment. (Applicable to cast iron beams.)

The combinations of those simple methods all fulfil the condition that the product of the depth and area, or the product of the breadth and square of the depth, at each cross-section is to be proportional to the bending moment.

The following Figures and Table illustrate the application of those principles to cast iron beams:—



Mode of Loading and Supporting.	δA^2 proportional to	Depth A constant; Figure of Horizontal Section.	Breadth b constant; Figure of Vertical Longitudinal Section.
I. Fixed at A, loaded at B.	Distance from B.	Triangle, apex at B, Fig. 14 a.	Parabola, vertex at B, Fig. 14 b.
II. Fixed at A, uniformly loaded.	Square of distance from B.	Pair of parabolas, vertices touching each other at B, Fig. 14 c.	Triangle, apex at B, Fig. 14 d.
III. Supported at A and B, loaded at C.	Distance from adjacent point of support.	Pair of triangles, common base at C, apices at A and B, Fig. 14 e.	Pair of parabolas, vertices at A and B, meeting at C, Fig. 14 f.
IV. Supported at A and B, uniformly loaded.	Product of distances from points of support.	Pair of parabolas, vertices at C, in middle of beam; common base A B, Fig. 14 g.	Ellipse A D B, Fig. 14 h.

One combination only needs to be specified—viz.,

METHOD IV.—In pieces having a solid circular cross-section, make the diameter proportional to the cube root of the bending moment. (Applicable to timber masts, yards, &c.)

49. Deflection of Beams.—The deflection or change of figure which a beam undergoes through the action of a transverse load, is the combined effect of the bending moments and racking forces exerted on the various cross-sections of the beam. In most cases which occur in practice, the part of the deflection due to the racking forces is so small, that it may either be neglected, or allowed for by an approximate process after the deflection due to the bending moments has been computed.

Supposing, then, in the first place, that the whole sensible deflection is produced by the bending moments, the following is the process for determining it:—

RULE I.—To find the radius of curvature at a given point in the originally straight longitudinal axis of the beam: multiply the moment of inertia of the cross-section* (Article 46) by the modulus of elasticity, and divide by the bending moment.

RULE II.—To find the deflection: having divided the length of the beam into a sufficient number of intervals, and computed the radii of curvature at the middle points of those intervals by Rule I., draw a curve (A T B, Fig. 15), made up of short circular arcs of the given lengths (A I, I 2, 2 3, &c.), and radii (the centres of curvature being O₁, O₂, O₃, &c.); this curve will represent the bent longitudinal axis of the beam. If any interval is free from bending



action, it is to be represented by a straight line. Then proceed as follows:—If the positions of two points, A and B, are fixed, draw the straight chord, A B; the greatest perpendicular distance, T V, of the curve from that chord will be the deflection required; but if the beam is made fast so as to be fixed in direction at some point, such as T,

and A is its projecting end, draw A V parallel, and T V perpendicular, to a tangent to the curve at T, cutting each other in V; T V will be the deflection required.

* The moments of inertia of similar cross-sections are proportional to their breadths and the cubes of their depths; also to their areas and the squares of their depths. Hence the moment of inertia, I, of a cross-section of the breadth b, depth h, and area S, may be computed by either of the two formulæ—

$$I = \pi' b h^3, \text{ or } I = q' S h^2;$$

π' and q' being multipliers depending on the form of section, whose values for ordinary forms are given by the following rule:—

Multiply the multiplier in the Table of Article 46 (π or q as the case may be), by the ratio in which the distance of the most severely strained layer from the neutral axis is less than the depth; the product will be the multiplier required (π' or q' as the case may be).

In algebraical symbols—

$$\pi' = \frac{\pi h_1}{h}; \quad q' = \frac{q h_1}{h}.$$

For Cases I. to VIII. inclusive of the Table referred to, the ratio $\frac{h_1}{h} = \frac{1}{2}$.

The radii of curvature are in general so long, that they cannot be conveniently contained within the drawing, if laid down to the same scale with the lengths of the arcs. In such cases, an exaggerated drawing is to be made, by laying down the radii upon a scale as many times smaller than the scale for lengths of arcs as may be convenient; the result being, that the deflection is magnified in the same proportion in which the radii are diminished: for example, if the radii are diminished to one-hundredth of their true proportionate lengths, as compared with the span of the beam, the deflection is magnified one hundred fold.†

The following rule is useful in comparing together the deflections of beams, or deducing the deflection of one beam from that of another:—

RULE III.—The deflections of similar beams, similarly loaded and supported, are directly as the loads and cubes of the lengths, and inversely as the moments of inertia of the cross-sections, and the stiffness of the materials.

The preceding rules relate to the deflection of beams under any given load, not exceeding the proof load.

The most important deflection of a beam in practice is that which is produced by the proof load itself, and which depends on the following principles. If we denote by proof curvature the sharpest curvature which a beam can bear without danger at a given cross-section, the radius of that curvature is found as follows:—

RULE IV.—As the sum of the intensities of stress at the stretched and compressed sides of the beam when under the proof load

- : is to the modulus of elasticity,
- :: so is the depth of the cross-section
- : to the radius of proof curvature.

This being taken as the length of the shortest radius of curvature, the radii of curvature at a series of other points are to be computed from their proportions to the shortest, agreeably to Rule I., and then the deflection may be found by Rule II.

RULE V.—The proof deflections of similar beams, similarly supported and loaded, are directly as the squares of their lengths, and inversely as their radii of proof curvature.

By the aid of such Tables as that annexed, Rules III. and V. are brought into the following forms, which are convenient for practical use.

RULE VI.—To find the deflection of a beam under ANY LOAD not exceeding the proof load: multiply the load by the cube of the

† The algebraical formulæ corresponding to these processes are as follows:—

Let co-ordinates be measured from a point in the longitudinal axis of the beam whose position is fixed by the mode of support; and let v according as it is {positive } represent the {elevation } of any other point in that axis, at the distance, x , from the origin of co-ordinates. Let $i_0 = \frac{dv}{dx_0}$ represent the slope at the origin, which, if the beam is simply supported there, is for the present an unknown quantity; but if the beam is fixed so as to be horizontal there, is = 0. Let $i = \frac{dv}{dx}$ denote the slope at any other point, and r the radius of curvature. Then that radius is computed by the formulæ—

$$r = \frac{M}{E i} \quad \dots \dots \dots \text{(Rule I.)}$$

and in the cases which occur in practice, we have, very nearly—

$$v = \int i dx = \int \int \frac{dx}{r} + i_0 x; \quad \dots \dots \dots \text{(Rule II.)}$$

When $i_0 = 0$, as in a beam fixed in direction as well as supported at one end, the last equation becomes simply—

$$v = \int \int \frac{dx^2}{r};$$

and completes the solution of the problem; but when i_0 is unknown, a further process is necessary: for example, in a beam supported at the two ends, the value of v for the end furthest from the origin is to be put = 0, and the value of i_0 deduced from the equation so formed; and then the maximum value of v is to be found; that is to say, its value at the point where $i = 0$.

length, and by a multiplier depending on the manner of loading and supporting, and divide by the moment of inertia of the greatest cross-section, and the modulus of elasticity.

RULE VII.—To find the PROOF DEFLECTION: multiply the square of the length by a multiplier depending on the manner of loading and supporting, and divide by the radius of proof curvature, as found by Rule IV.*

Case.	Multipliers for Deflection.	
	Proof Load. n''	Any Less Load. n'''
A. UNIFORM CROSS-SECTION.		
I. Constant Moment of Flexure,.....	$\frac{1}{8}$	
II. Fixed at one end, loaded at the other,.....	$\frac{1}{8}$	$\frac{1}{8}$
III. Fixed at one end, uniformly loaded,.....	$\frac{1}{8}$	$\frac{1}{8}$
IV. Supported at both ends, loaded in the middle,...	$\frac{1}{8}$	$\frac{1}{8}$
V. Supported at both ends, uniformly loaded,.....	$\frac{1}{8}$	$\frac{1}{8}$
B. UNIFORM STRENGTH AND UNIFORM DEPTH. (The curvature of these is uniform.)		
VI. Fixed at one end, loaded at the other,.....	$\frac{1}{8}$	$\frac{1}{8}$
VII. Fixed at one end, uniformly loaded,.....	$\frac{1}{8}$	$\frac{1}{8}$
VIII. Supported at both ends, loaded in the middle,...	$\frac{1}{8}$	$\frac{1}{8}$
IX. Supported at both ends, uniformly loaded,.....	$\frac{1}{8}$	$\frac{1}{8}$
C. UNIFORM STRENGTH AND UNIFORM BREADTH.		
X. Fixed at one end, loaded at the other,.....	$\frac{8}{9}$	$\frac{8}{9}$
XI. Fixed at one end, uniformly loaded,.....	1	$\frac{1}{2}$
XII. Supported at both ends, loaded in the middle,...	$\frac{1}{2}$	$\frac{1}{2}$
XIII. Supported at both ends, uniformly loaded,.....	0.1427	0.01784

One of the chief practical uses of a knowledge of the laws of the proof deflection of a beam is, to fix beforehand the proportion of the depth of a beam to its length, so that the proof deflection shall bear a given proportion to the length, which is done by the following rule:—

RULE VIII.—Multiply the proportion in which the length is to be greater than the proof deflection by the sum of the greatest intensities of tension and thrust under the proof load, and by the multiplier (n'') suited to the way in which the beam is to be loaded and supported; divide by the modulus of elasticity; the quotient will be the ratio in which the depth is to be less than the length.†

EXAMPLE.—Suppose, for a beam of wrought iron of uniform cross-section supported at the ends and uniformly loaded, that the moduli to be used in the calculation are—

	Lbs. on the square inch.
Greatest tension,.....	20,000
Greatest thrust,.....	12,000
Modulus of elasticity,.....	25,000,000

and that the proof deflection is to be $\frac{1}{80}$ of the length.

The multiplier, n'', from Case V. in the Table, is $\frac{5}{8}$. Consequently—

$$\frac{\text{Depth}}{\text{Length}} = \frac{5 \times 32,000 \times 600}{48 \times 25,000,000} = \frac{1}{12\frac{1}{2}} = 0.08.$$

Such is the method by which the depth of a beam is usually fixed, before proceeding to compute its breadth or its sectional area by Rules V. and VI. of Article 46.

50. *Deflection due to Racking.*—The proportion which the additional deflection produced by the racking forces bears to the deflection produced by the bending moments, may be found nearly enough for practical purposes as follows:—

* In algebraical symbols, let n'' and n''' be suitable multipliers;

$$\text{Deflection under any load} = \frac{n'' W l^3}{EI} \quad \text{(Rule VI.)}$$

$$\text{Radius of proof curvature} = \frac{Eh}{\frac{1}{8} + \frac{1}{8}} \quad \text{(Rule IV.)}$$

$$\text{Proof deflection} = \frac{n'' (\frac{1}{8} + \frac{1}{8}) l^3}{Eh} \quad \text{(Rule VII.)}$$

† Algebraically, let v be the proof deflection; then—

$$\frac{h}{l} = \frac{n'' (\frac{1}{8} + \frac{1}{8}) l^3}{E v}$$

By the Rules or the Table of Article 47, find the ratio in which the greatest racking stress on a cross-section of the beam is greater than the mean racking stress. Multiply that ratio by the ratio in which the direct elasticity is greater than the rigidity, by the square of the ratio in which the depth is less than the length, and by the constant $\frac{2}{3}$ for beams supported at the ends and uniformly loaded, or $\frac{1}{2}$ for beams supported at the ends and loaded in the middle; the product will be the proportion required.‡

EXAMPLE I.—In a uniformly-loaded I-shaped wrought iron beam, let the proportions be the same as in the example of Article 47, so that the greatest racking stress exceeds the mean in the ratio of 3.2 : 1; also let the direct elasticity be three times the rigidity, and the length $12\frac{1}{2}$ times the depth; then the proportionate addition to the deflection through racking is—

$$\frac{2}{3} \times 3.2 \times 3 \times (0.08)^2 = .0246 \text{ nearly, or less than } \frac{1}{40}.$$

EXAMPLE II.—Let a pine beam be rectangular, so that the greatest racking stress is $\frac{3}{2} \times$ the mean; let it be supported at the ends and loaded in the middle; let its direct elasticity be 20 times its rigidity, and let the span be 16 times the depth; then the proportionate increase of deflection through racking is—

$$\frac{1}{2} \times \frac{3}{2} \times 20 \times \frac{1}{16^2} = .0586 \text{ nearly, or about } \frac{1}{17}.$$

51. The *Resilience* or *Spring* of a Beam means, the quantity of mechanical work required in order to produce the proof deflection of the beam. In a beam loaded at one point only, it is equal to the proof load multiplied by half the proof deflection at the loaded point. When the load is distributed, each part of the load is to be multiplied by half the proof deflection produced at its own point of application, and the products added together. For beams of the same material and of similar form, similarly loaded and supported, the resilience is proportional to the volume of the beam. For different materials it varies as the modulus of resilience; as to which, see Article 34 of this Division.

52. *Allowance for the Weight of a Beam.*—When a beam is of great span, its own weight may bear a proportion to the load which it has to carry, sufficiently great to require to be taken into account in determining the dimensions of the beam. The following is the process to be performed for that purpose:—Assume, as a *provisional load*, the external load upon the beam only, and find a *provisional breaking load* by the use of proper factors of safety. Then, by Rule V. or Rule VI. of Article 46, find a *provisional breadth* or a *provisional area* suited to the *provisional breaking load*, and thence compute the *provisional weight* of the beam.

Multiply the provisional weight so found by a proper factor of safety, and subtract the product from the provisional breaking load. With the remainder as a divisor, divide the provisional breaking load; the quotient will be a number greater than unity, and will be the factor by which the provisional breadth or area, and the provisional weight, are to be multiplied, in order to find the *true breadth* or *true area*, and the *true weight*, of the beam.

53. *Continuous Beams* is the term applied to those beams which either rest on more than two supports, or are fixed in direction as well as supported at one or both of two points of support; the

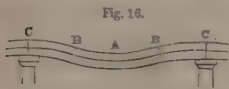
‡ Let v be the deflection due to the bending moments; v', the additional deflection due to the racking forces; $\frac{e_0 S}{F}$, the ratio so expressed in Article 47; E, the direct elasticity; C, the rigidity; l, the length;

h, the depth; then, approximately, for uniformly-loaded beams—

$$\frac{v'}{v} = \frac{2}{5} \cdot \frac{e_0 S}{F} \cdot \frac{E}{C} \cdot \frac{l^2}{h^2}$$

For beams loaded in the middle, put $\frac{2}{5}$ instead of $\frac{2}{3}$ (see Rankine "On Applied Mechanics," p. 342).

effect being to increase the resistance to bending and to cross-breaking. Fig. 16 represents the general character of the figure assumed by one span of a continuous beam when bent by a load.



At A, the middle of the span, the beam becomes concave upwards, like an ordinary beam; at the points of support, C, C, it becomes convex upwards, and the bending moment, instead of being = 0, is expressed by a *negative quantity*; at a pair of intermediate points, B, B, called *points of contrary flexure*, the bending moment is = 0, and the direction of curvature changes. The greatest bending moment occurs in some cases at A, and in others, at C, C, the latter being the more frequent.

The particular problems respecting beams, which have been treated of in the preceding Articles, have all reference to cases in which the determination of the racking force and bending moment at each point, and of the curvature, slope, and deflection, are simple and direct processes, proceeding step by step from the determination of one quantity to that of another. In the present Article, cases are considered in which the racking force and bending moment depend, to a greater or less extent, on the curvature, slope, and deflection; and in which, consequently, the algebraical process of elimination is often required, two or more unknown quantities having to be determined at once by solving an equal number of equations at the same time. It would be out of place in the present treatise to give the details of such investigations: the most generally useful results are the following:—

CASE I.—*Beam of Uniform Section, firmly fixed at both ends, loaded in the middle.*

Bending moment at A = contrary bending moment at C, C = $\frac{1}{2}$ bending moment at middle of ordinary beam = $\frac{\text{load} \times \text{span}}{8}$;

Proof deflection = $\frac{1}{2}$ proof deflection of ordinary beam (see Art. 49).

Thus the strength and stiffness of a given beam, in this case, are doubled by fixing the ends.

The points of contrary flexure are midway between the middle and the ends.

CASE II.—*Beam of Uniform Section, firmly fixed at both ends, loaded uniformly distributed.*

Bending moment at A = $\frac{1}{3}$ bending moment at middle of ordinary beam = $\frac{\text{load} \times \text{span}}{24}$;

Contrary bending moment at C, C = $\frac{2}{3}$ bending moment at middle of ordinary beam = $\frac{\text{load} \times \text{span}}{12}$;

Proof deflection = $\frac{5}{16}$ proof deflection of ordinary beam (see Article 49).

Thus the strength of a given beam, in this case, is increased $1\frac{1}{2}$, and its stiffness $3\frac{1}{2}$ times, by fixing the ends.

The points of contrary flexure are at 0.289 of the span from the middle.

54. *Bending combined with Longitudinal Stress.*—In every case in which the line of action of the load upon a beam is not exactly parallel to a given cross-section of that beam, that cross-section sustains a direct longitudinal stress, tensile or compressive as the case may be, in combination with the stress due to the bending and

racking actions. The *amount* of the longitudinal load so produced is equal to the component of the entire load in a direction perpendicular to the given cross-section (see Div. I., Article 54); and the *intensity* of the stress produced by it is uniform, and is found by dividing its amount by the area of the cross-section. The resultant longitudinal stress on each particle is the sum or difference of the longitudinal stress due to the bending action and of that due to the longitudinal component of the load, according as they are of the same or of contrary kinds.

For example, Fig. 17 represents a “davit” or small crane, made fast at the points C and B, and loaded at A with a force represented by \overline{AP} . To find the straining actions which take place at a given cross-section, such as S, draw AN and PF perpendicular, and AF and PN parallel, to the plane of that cross-section; then \overline{AF} will represent the racking force, and \overline{AN} the amount of the *direct longitudinal load*, so that the intensity of the *direct longitudinal stress* will be—

$$\frac{\overline{AN}}{\text{Area } S'}$$

From the neutral axis of the section S let fall SL perpendicular to AP; then—

$$\overline{AP} \times \overline{SL}$$

will be the bending moment at S.

In designing beams to bear such combined straining actions, it is convenient to use the following—

RULE.—*Multiply the longitudinal component of the load by the depth of the beam, and by the multiplier denoted by q, in the Table of Article 46 of this Division: divide the product by the bending moment; the quotient will be the proportionate fraction by which the strength of the beam should be increased, beyond what is necessary to bear the bending action alone.*

EXAMPLE.—Suppose load = 4 tons;
Angle which it makes with the given cross-section,
30°; consequently longitudinal component = $4 \times$
 $\sin. 30^\circ =$ 2 tons;
Leverage of load 50 inches; therefore bending
moment = $4 \times 50 =$ 200 inch-tons;
Depth of beam 12 inches; section I-shaped, and so proportioned
that $q = \frac{1}{3}$; then—

$$\frac{2 \times 12 \times \frac{1}{3}}{200} = \frac{8}{200} = .04;$$

and the beam must be made stronger than would be required to resist the bending moment alone, in the proportion of 1.04 to 1, or of 26 to 25.

The same principles are applicable when a strut or tie is subjected to a load whose line of action is parallel to, but does not coincide with, the *longitudinal axis* of the strut or tie; that is, the line traversing the centres of all its cross-sections. In this case, the whole load produces direct longitudinal stress, with which is combined the stress due to a bending moment found by multiplying the load by the perpendicular distance of its line of action from the longitudinal axis of the strut or tie.

* For details, see Rankine “On Applied Mechanics,” p. 332; “On Civil Engineering,” p. 282.

SECTION VII.—OF TWISTING AND WRENCHING.

55. *Moment of Torsion, or Twisting Moment*, is the name applied to the moment of a pair of equal and opposite couples of forces applied to the two ends of a bar, in planes perpendicular to its axis, so as to balance each other, and twist the bar. Such is the condition of strain of a shaft or axle through which power is transmitted.

If the bar be conceived to be divided into fibres, originally straight and parallel to its axis, each of those fibres, in the twisted state of the bar, becomes a portion of a screw; and each particle of the bar is racked or distorted, and exerts a racking stress.

56. *Resistance to Twisting*.—If a bar under the straining action of a twisting couple be conceived to be divided by a cross-section perpendicular to its axis, the particles traversed by that cross-section exert a *moment of resistance to twisting*, which is the sum of the moments of their several racking stresses relatively to the axis of the bar; and which, in cylindrical and square bars, has the values given by the following rules:—

RULE I.—If the bar is solid, divide the angle through which the bar is twisted, in circular measure (Div. I., Art. 30), by the length of the bar; multiply the quotient by the fourth power of the thickness or diameter of the bar, the modulus of rigidity of the material (Div. III. Art. 42), and a numerical factor, whose value is—

For round bars,.....	0.098, or $\frac{1}{10}$ nearly;
For square bars,.....	0.1405, or $\frac{1}{7}$ nearly;

the product will be the moment of resistance corresponding to the given angle.

RULE II.—If the bar is hollow, substitute, in Rule I., the difference of the fourth powers of the inside and outside diameters for the fourth power of the outside diameter.*

57. *Resistance to Wrenching*.—The ultimate resistance to wrenching, or wrenching moment, is the twisting moment which is just sufficient to wrench the bar asunder; the *proof* resistance is the utmost twisting moment which the bar can bear without injury at a single trial; the *working* resistance, that which it is considered safe to subject it to during its ordinary use.

For round and square bars, the wrenching moment may be found by the following Rules:—

RULE I.—If the bar is solid, multiply the cube of the diameter or thickness by the modulus of wrenching (which is the same nearly with the modulus of resistance to shearing; Article 43) and by a numerical factor, whose values are—

For round bars,.....	0.196, or $\frac{1}{5}$ nearly;
For square bars,.....	0.281, or $\frac{2}{7}$ nearly;

the product will be the moment required.

RULE II.—If the bar is hollow, divide the difference of the fourth powers of the outside and inside diameters by the outside diameter, and use the quotient in Rule I., instead of the cube of the outside diameter.†

* Let A be the outside and A' the inside diameter of a round bar; or $A \times A'$ the outside and $A' \times A'$ the inside dimensions of a square bar; l , the length of the bar; θ , its angle of torsion in circular measure; C , the modulus of rigidity of the material; k , a factor = .098 for round and .1405 for square bars; then—

$$\text{Moment of Resistance} = \frac{C k \theta (A^4 - A'^4)}{l}$$

† Let A be the outside diameter or thickness; A' the inside diameter; f , the modulus of resistance to wrenching; k , the factor whose values have been given in the preceding note; then—

$$\text{Wrenching moment} = 2 k f \frac{A^4 - A'^4}{A}$$

The following rule is in many cases convenient:—

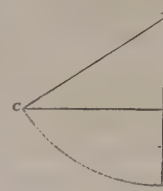
RULE III.—The resistance of a round, elliptical, or flat bar to wrenching, is nearly double of its resistance to breaking across in the weakest direction (Article 46).

RULE IV.—To find the transverse dimensions of a round or square solid bar whose wrenching moment is given; divide the given moment by the modulus of wrenching, and by the factor 0.196 for a round bar, or 0.281 for a square bar; the cube root of the quotient will be the required diameter or thickness.

58. *Bending and Twisting combined*.—

When a bar is subjected at the same time to a bending moment and a twisting moment (as crank-shafts often are), draw AB , in Fig. 18, to represent the bending moment, and BC perpendicular to AB to represent the twisting moment; in AB produced take $AD = AC$, and bisect BD in E ; the greatest stress on the particles will be the same with that due to a bending moment represented by AE .‡

Fig. 18.



SECTION VIII.—OF JOINTS AND FASTENINGS IN GENERAL.

59. *General Explanations*.—The particular kinds of joints and fastenings which occur in shipbuilding, will be considered in detail in the sequel of the present Division and in the Fourth Division. The object of the present Section is to explain some principles which are applicable to joints and fastenings generally; *joints* being the surfaces at which the pieces of a frame or other structure touch each other; and *fastenings*, the auxiliary pieces by means of which the principal pieces are connected together at those joints.

Joints may be classed as follows:—

- I. Joints for lengthening ties.
- II. Joints for lengthening struts.
- III. Joints for lengthening beams.
- IV. Joints for supporting beams on beams.
- V. Joints for supporting beams on pillars.
- VI. Joints for connecting struts with ties.

But of those classes the first three alone require special mention in this Section.

Fastenings may be classed as follows:—

I. Pins; including treenails, nails, spikes, screws, bolts, and keys: being fastenings which are exposed principally to racking stress.

II. Straps and tie-bars, including covering-straps and fish-pieces of ties, being fastenings which are exposed principally to tension.

III. Knees and fish-pieces exposed principally to bending.

IV. Sockets and coaks, for keeping the ends or joints of pieces in their places.

In designing and executing all kinds of joints and fastenings, the following general principles are to be adhered to as closely as may be practicable:—

I. To cut the joints and arrange the fastenings so as to weaken the pieces that they connect as little as possible.

II. To place each abutting surface in a joint as nearly as possible perpendicular to the pressure which it has to transmit.

‡ AE may be found by calculation thus:—

$$AE = \frac{AB}{2} + \sqrt{\frac{AB^2}{4} + BC^2}$$

III. To proportion the area of each such surface to the pressure which it has to bear, so that the material may be safe against injury under the heaviest load which occurs in practice; and to form and fit every pair of such surfaces accurately, in order to distribute the stress uniformly.

IV. To proportion the fastenings so that they may be of equal strength with the pieces which they connect.

V. To place the fastenings in each piece so that there shall be sufficient resistance to the giving way of the joint by the fastenings shearing or crushing their way through the material of the piece.

Fastenings also differ in construction and arrangement, according to the nature of their own material, and of that of the pieces which they connect.

60. *Fastenings for Metal—Rivets, Keys, Bolts, &c.*—Rivets are made of the most tough and ductile iron. (See "Rivet Iron," in the Tables of Strength.) In order that a rivet may connect two or more layers of plates or flat bars firmly, and in order that the racking stress brought to bear on the rivet by a force tending to pull the plates asunder may be uniformly distributed throughout the sectional area of the rivet, it is essential that the rivet should fit its hole as tightly as possible (see Art. 43 of this Division). The longitudinal compression to which the rivet is subjected during the formation of its head, whether by hand or by machinery, tends to produce that result.

The ordinary dimensions of rivets in practice are as follows:—

Diameter of a rivet for plates less than half an inch thick, from once and a half to double the thickness of the plate.

For plates of half an inch thick and upwards, from once and a tenth to once and a half the thickness of the plate.

Length of a rivet before clenching, measuring from the head = sum of the thickness of the plates to be connected + $2\frac{1}{2}$ times the diameter of the rivet.

Inasmuch as the effective resistance of rivets to shearing, after deducting an allowance for the effect of their not fitting absolutely tight, is nearly the same with the tenacity of good iron plates, the distance apart of the rivets used to connect two pieces of plate iron together is regulated by the principle that *the joint sectional area of the rivets should be equal to the sectional area of plate left after punching or drilling the rivet holes*. That principle leads to the following rule—

RULE I.—To find the *pitch*, or distance from centre to centre of the rivets in one row: *multiply the square of the diameter of a rivet by .7854, and by the number of rows of rivets in the joint, and divide by the thickness of the plates; to the quotient add the diameter of a rivet*.

Each plate is weakened by the rivet holes in the ratio which the breadth of metal left between each pair of holes in one row bears to the pitch of the holes.

In single-riveted joints there is but one row of rivets: in double-riveted joints there are two rows, which form a zig-zag; in "chain-riveted" joints there may be any number of rows. As to the strength of single and double riveted joints, see Article 30 of this Division.

Suppose that in a chain-riveted joint the pitch of the rivets is fixed, so as not to weaken the plates below a given limit; then in order to find how many rows of rivets there should be—in other words, how many rivets there should be in each file—the following rule may be used:—

RULE II.—*Multiply the thickness of the plate by the breadth of solid metal left between a pair of rivet-holes, and divide by the*

sectional area of a rivet; that is, by $.7854 \times$ the square of its diameter.

Pins, Keys, and Wedges are, like rivets, themselves exposed to a racking stress, while they serve to transmit a pull or thrust from one piece of an iron frame to another; and the rule for determining their proper sectional area is the same.

In order that a wedge or key may be safe against slipping out of its seat, its angle of obliquity ought not to exceed the angle of repose of iron upon iron, which, to provide for the contingency of the surfaces being greasy, may be taken at about 4° .

If a *bolt* has to withstand a racking stress, its diameter is to be determined like that of a cylindrical pin. If it has to withstand tension, its diameter is to be determined by having regard to its tenacity. In either case the effective diameter of the bolt is its least diameter; that is, if it has a screw on it, the diameter of the spindle inside the thread.

The projection of the thread is usually *one-half of its pitch*; and the pitch, or distance between two adjoining turns of the thread, should not in general be greater than *one-fifth of the effective diameter*, and may be considerably less.

In order that the resistance of a screw or screw-bolt to rupture by stripping the thread may be at least equal to its resistance to direct tearing asunder, the length of the nut should be *at least one-half of the effective diameter of the screw*; and it is often in practice considerably greater; for example, once and a half that diameter.

The head of a bolt is usually about twice the diameter of the spindle, and of a thickness which is usually greater than five-eighths of that diameter.

61. *Fastenings for Timber—Treenails, Coaks, Spikes, Nails, Screws, Bolts, Straps, Knees, &c.*—Wooden pins, used as fastenings for timber, when of large diameter, are known as *treenails*. From experiments by Mr. Parsons it appears, that the ultimate shearing stress of *British oak treenails* is about 4000 lbs. per square inch of section; and that in order to realize that resistance, the thickness of the planks fastened by the treenails ought to be about *three times the diameter of a treenail*. Further explanations as to the use of treenails will be given in the sequel. *Coaks*, or *Dowels*, are small blocks of some hard wood, fitting into holes in the abutting ends or joints of two pieces of timber, in order to keep them in their proper relative position.

Metal fastenings for timber are usually of iron, copper, or some alloy of copper. Some sorts of timber, and especially oak, contain juices which corrode iron; and sea-water rapidly corrodes iron when that metal is in contact with a less oxidable metal, such as copper; and hence the advantage of using copper rather than iron fastenings in wooden ships, and especially in those parts which are sheathed with copper, notwithstanding the smaller strength and greater cost of copper fastenings. The following are the ordinary kinds of metal fastenings for timber:—

I. *Nails and Spikes*.—Where nails are exposed to any considerable strain those made by hand should be used, as they are stronger than those made by machinery.

The weight in lbs. of a thousand of the "flooring brads" commonly used in carpentry, may be roughly computed by taking *twice the square of their length in inches*.

The nails or spikes used for fastening planks to beams are usually of a length equal to from twice to twice and a half the thickness of the planks.

The following are the results, as stated by Tredgold, of experiments by Bevan on the force required to draw nails of different sizes out of *Dry Christiania Deal*, into which they had been driven to different depths *across the grain* :—

Kind of Nails.	Length. Inches.	No. to the Lb.	Inches driven.	Force to draw. Lbs.
Sprigs,.....	0·44	4,560	0·4	22
"	0·53	3,200	0·44	37
Threepenny brads,.....	1·25	618	0·50	58
Cast iron nails,.....	1·00	380	0·50	72
Fivepenny nails,.....	2·00	139	1·50	320
Sixpenny nails,.....	2·50	73	1·00	187
"	2·50	73	1·50	327
"	2·50	73	2·00	530

So far as these results can be expressed by a general law, they seem to indicate that the force required to draw a nail, driven across the grain of a given sort of wood, varies nearly as the *cube of the square root of the depth to which it is driven*; and that it increases with the diameter of the nail, but in a manner which has not yet been expressed by a mathematical law.

The following are the results of Bevan's experiments on the force required to draw a "sixpenny nail" of 73 to the lb., which had been driven one inch into different sorts of timber :—

	Lbs.
Deal, across the grain,.....	187
Oak, "	507
Elm, "	327
Beech, "	667
Green Sycamore, "	312
Deal, endwise,.....	87
Elm, "	237

The following were the forces required to draw asunder a pair of planks joined by *two nails* of 73 to the lb. :—

	Lbs.
Deal $\frac{3}{4}$ inch thick,.....	712
Oak 1 inch thick,.....	1009
Ash 1 inch thick,.....	1420

II. *Screws*.—The holding power of screw-nails, or wood-screws, is probably proportional nearly to the product of the diameter of the screw, and of the depth to which it is screwed into the wood. The following are the results of Bevan's experiments, quoted by Tredgold, on the force required to draw screws out of planks of *half an inch thick*, the screws being 0·22 inch in diameter over all, and 0·035 inch in depth of thread, with 12 threads to the inch :—

	Lbs.
Beech,.....	460 to 390
Ash,.....	790
Oak,.....	760
Mahogany,.....	770
Elm,.....	665
Sycamore,.....	830

III. *Bolts*.—The rules for proportioning bolts which have to withstand a racking stress have already been stated.

The sides of a piece of timber are protected against the crushing action of the head and nut of a bolt by means of flat rings called "*washers*;" the area of each washer being at least as many times greater than the sectional area of the bolt as the tenacity of the bolt is greater than the resistance of the timber to crushing; that is to say, for fir the diameter of the washer may be made about $3\frac{1}{2}$ times the diameter of the bolt, and for oak about $2\frac{1}{2}$ times.

When a bolt is oblique to the piece of timber that it traverses, the timber may either have a notch cut in it with a surface

perpendicular to the bolt, to bear the pressure of the washer, or it may be notched to receive a bevelled washer, one of whose surfaces fits the notch in the wood, while another being perpendicular to the axis of the bolt, bears the pressure of the nut or head, as the case may be.

The screws of bolts are usually made of the following proportions, or nearly so:—the depth of the thread one-tenth, and the pitch one-fifth, of the internal diameter. A bolt which has to be often removed may be made fast by having a slot or oblong hole in one of its ends, through which a key or wedge is driven.

IV. *Straps* are used to bind pieces of timber together. According to the usual proportions of straps the breadth ranges from four times to eight times the thickness. When a strap has eyes in its ends, for bolting them to the sides of a beam, it ought to be either broadened or thickened round each eye, so that the sectional area of the iron may be at least as great at the sides of the eye as in other parts of the strap. When a strap is to embrace completely a piece or pieces of timber, it may, when practicable, be welded into a rectangular hoop, and driven on from one end of the timber; but when that is impracticable or inconvenient, it must be made with screws on its ends, of the same sectional area with its flat part, upon which screws a cross-piece is to be made fast with nuts.

V. *Iron Sockets*, shaped to fit the ends of pieces of timber, furnish a convenient means of making various joints in framework, especially at points where struts meet each other, or have to be connected with tie-rods. If thrust alone is to be borne by the socket, cast iron is the most convenient material; if any considerable tension is to be borne, strong wrought iron plates are best.

62. *Joints for lengthening Metal Ties*.—A metal tie may consist either of one bar, or of several bars side by side, or of wires lying parallel in a bundle or spun into a rope; it may be in one length, or in two or more lengths joined together; if the lengths are numerous and short, they become *links*, and the whole tie a chain.

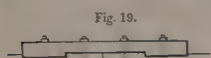
Plate Iron Ties.—The best mode of joining two lengths of a plate iron tie is by means of a rivetted butt-joint, with fish-plates or covering straps. The principles according to which the dimensions, number, and arrangement of the rivets are to be determined have been explained in Article 60 of this Section; and it has there also been shown to what extent the *effective sectional area* of the tie is diminished by the rivet holes, so as to be less than the total sectional area. The holes are more correctly made by drilling than by punching. Punched holes are a little smaller at the end where the punch enters than at the other end; and when two punched plates are to be rivetted together, the smaller ends of the holes ought to be placed together, in order to make a good joint.

When a plate iron tie is built of several layers, they should *break joint* with each other; and at each joint there should be either a covering strip or a pair of covering strips, to transmit that share of the tension which belongs to the layer of plates in which the joint occurs.

Tie-rods or *Tie-bars* may be round, square, or flat, and may be made fast at the ends by pins passing through round eyes, by wedges driven into oval eyes or slots, or by screws and nuts. The proportions of these fastenings have been considered in Article 60 of this Section. Wedges and screws admit of being used to tighten the tie.

When an *eye* is formed on the end of a tie-bar, care should be taken that the sides of the eye are of sufficient strength. The tension is not uniformly distributed in them, being more intense at the inner side than at the outer. To allow for this, the sectional area may be made one-half greater than would be necessary if the tension were uniformly distributed.

63. *Joints for lengthening Timber Ties.*—Lengthening timber ties is performed by *fishing* or by *scarfing*. In a fished joint the two



pieces of the tie abut end to end, and are connected together by means of "fish-pieces" of wood or iron which are bolted to their sides; in a scarfed joint the ends of the two pieces of the tie overlap each other. Fig. 19 is a fished joint; Figs. 20, 21, and 22 are called scarfs;

Fig. 20.



Fig. 21.

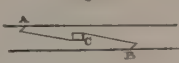


Fig. 22.



though in Figs. 20 and 21 the ties are in fact fished with iron as well as scarfed.

In a *plain fished joint* the fish-pieces have plane surfaces next the tie, so that the connection between them and the tie for the transmission of tension depends wholly on the strength of the bolts, together with the friction which they may cause by pressing the fish-pieces against the sides of the tie. The tie is only weakened so far as its effective sectional area is diminished by the bolt-holes. The joint sectional area of the fish-pieces should be equal to that of the tie. The joint sectional area of the bolts should be at least *one-fifth* of that of the timber left after cutting the bolt-holes; and the bolts should be square rather than round. The bolt-holes should be so distributed, and placed at such distances from the ends of the two parts of the tie, that the joint area of both sides of the layer of fibres, which must be shorn out of one piece of the tie before the bolts can be torn out of its end, shall be as much greater than the effective area of the tie as the tenacity of the wood is greater than its resistance to shearing; as to which proportion, see the Tables. The same rule regulates the places of the bolt-holes in the fish-pieces.

The fish-pieces and the parts of the tie may also be connected by *indents*, as at the upper side of Fig. 19, or by *joggles* or *keys*, as at the lower side of the same figure. In either case, the effective area of the tie is reduced by the cutting of the indents or of the key-seats, at A and B. The area of abutting surface of the indents, or of the key-seats, should be sufficient to resist safely the greatest force to be exerted along the tie; and their distances from the ends of the fish-pieces and of the parts of the tie should be sufficient to resist safely the tendency of the same force to shear off two layers of fibres.

A timber tie may be fished with plates of iron, due regard being paid to the greater tenacity of the iron in fixing the proportions of the parts; and the iron fish-plates may be indented into the wood. Fig. 20 represents a joint in which the parts of the timber tie are scarfed together, and at the same time fished with iron plates, which are indented into the wood at the ends.

Fig. 21 represents a scarfed joint for a tie, which will hold without the aid of bolts or straps. At C is a key or joggle of some hard kind of wood, which is wedged in so as to tighten the joint moderately. The depth of the key is one-third of the depth of the beam. It is evident that this joint, as shown in the figure, has only one-third of the strength of the solid timber tie; but its strength may be considerably increased by bolting on iron fish-plates at A and B.

Fig. 22 shows a scarfed joint with several keys, which should all be driven equally tight. It is also fished with iron plates, indented into the wood at the ends.

The following practical rules are given by Tredgold for the proportion which the length of a scarf (between A and B in each of the figures) should bear to the depth of the tie:—

	Without Bolts.	With Bolts.	With Bolts and Indents.
Leaf-wood (as Oak, Ash, or Elm),.....	6	3	2
Pine-wood,.....	12	6	4

64. *Joints for lengthening Struts.*—At each joint in a post, pillar, or other strut, the two pieces should abut against each other at a plane surface, perpendicular to the direction of the thrust; and to keep them steady they may either be fished on all four sides, or have their abutting ends inclosed in a socket made to fit them or connected by a coak. Joints in struts ought, if possible, to be stayed laterally.

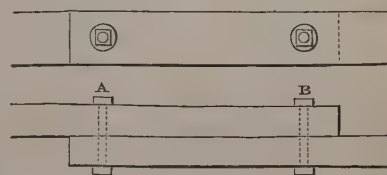
65. *Joints for lengthening Timber Beams* may be made either by fishing or by scarfing; and in either case the joints should as far as practicable be placed where the bending moment is small. The construction of the joints should be the same with that of joints for lengthening ties, with the following qualifications:—

I. At the compressed side of the beam, its two pieces should have a square abutment against each other; hence oblique surfaces, such as those in Fig. 21, are to be avoided.

II. The surfaces of the scarf ought to be parallel to the direction of the load (that is to say, in general, vertical: so that in Figs. 20 and 22, the plane of the paper shall represent a horizontal plane); for it was found, in experiments by Colonel Beaufoy, that a scarfed beam was stronger with the scarf "up and down" than "flatwise."

III. When two lengths of a beam, or other piece subjected to bending action, stand edgewise side by side overlapping each other, and are connected together at two points (as A and B, Fig. 23) by suitable fastenings, such as pins or straps, the amount

Fig. 23.



of the stress on each fastening is found by dividing the bending moment exerted on the beam at the given fastening by the distance \overline{AB} between the two fastenings.

TABLE OF PROPERTIES OF METALS AND MISCELLANEOUS SUBSTANCES.

METALS.	WEIGHT.		STRENGTH.				MODULUS OF ELASTICITY. Lb. on the square inch.	REFERENCES AND EXPLANATIONS.
	Specific gravity.	Lb. in a cubic foot.	TENSILE FORCE, lb. on the square inch.	SHEARING FORCE, lb. on the square inch.	CRUSHING FORCE, lb. on the square inch.	DEFORMING FORCE, lb. on the square inch.		
Aluminium,	2.70	168	18,000 Re.	27,700 R. Br. Du.	10,800 Re.	38,698	9,170,000 Mo.	REFERENCES TO AUTHORITIES. A., . . . Anderson. B., . . . Barlow (Peter). B. (P. W.), Barlow (P. W.). B. (W. H.), Barlow (W. H.). Be., . . . Bevan. Bel., . . . Beldor. Br., . . . Brown. Bru., . . . Brunel. Ca., . . . Carillion. Ch., . . . Chevardier. Cl., . . . Clay. Cu., . . . Cunningham. Co., . . . Couch. D., . . . Dobson. Do., . . . Doynes. Du., . . . Dunlop. Fa., . . . Fairbairn. Fi., . . . Fincham. Fo., . . . Fowke. Ga., . . . Galton. G., . . . Gordon. Gr., . . . Grassi. H., . . . Hodgkinson. J., . . . James. Ju., . . . Justen. K., . . . Kirkaldy. L., . . . Lamé. Ma., . . . Mallet. M., . . . Mayne. Me., . . . Mendis. Mo., . . . Morin. Mr., . . . More. M., . . . Muschenbroek. Ra., . . . Rankine. Re., . . . Rennie. Ro., . . . Rondelet. Sm., . . . Smith. Te., . . . Telford. Tr., . . . Tredgold. Tre., . . . Tresca. W., . . . Wertheim. Wi., . . . Wilmot. Y., . . . Young.
Aluminium Bronze.—See Bronze, Aluminium.							14,230,000 W.	
Brass—Cast,	8.399	524	18,000 Re.		10,800 Re.		9,873,000 Tr.	
Wire,	8.544	533	49,000 G.					
Bronze, or Gun-metal (copper 8, tin 1),	8.153	509	86,000 G.		132,000 A.			
Bronze, Aluminium (copper 9, aluminium 1),	7.68	480	73,000 A.					
Copper—Cast,	8.607	537	19,000 Re.					
Sheet—Rolled,	8.785	549	30,000					
Hammered,			33,600 Re.					
Bolts,			36,000 Fa.				17,067,000 W.	
Wire—Average,	8.879	554	60,000				14,000,000 Fa.	
Iron, Cast—Weak,	6.955 Fa.	434	13,400 H.		82,000 H.	33,000 Fa.	17,000,000	
Average,	7.100	443	16,500 H.		112,000 H.	40,000	22,900,000 H.	
Strong,	7.295 H.	455	29,000 H.		145,000 H.	43,500 H.		
IRON, CAST, VARIOUS KINDS (Fa. & H.).								
No. 1—Cold Blast,	from	12,694			56,455	36,698	14,000,000	
Hot,	to	17,466			80,561	39,771	15,380,000	
No. 2—Cold Blast,	from	13,434			72,193	29,889	11,539,000	
Hot,	to	16,125			88,741	35,916	15,510,000	
No. 3—Cold Blast,	from	13,348			68,532	33,453	12,586,000	
Hot,	to	18,855			102,408	39,600	17,036,000	
No. 4—Cold Blast,	from	13,505			82,734	28,917	12,259,000	
Hot,	to	17,407			102,030	38,394	16,301,000	
No. 5—Cold Blast,	from	14,200			76,900	35,881	14,281,000	
Hot,	to	15,508			115,400	47,061	22,508,000	
No. 6—Cold Blast,	from	15,273			101,831	35,640	15,852,000	
Hot,	to	23,468			104,884	43,497	22,733,000	
No. 7, Smelted by Coke without Sulphur,	from	23,461			129,876			
Toughened Cast-iron,	to	26,764			119,457			
No. 8, Hot Blast, after first melting,	from				98,500	39,690		
Hot,	to				163,744	56,080		
No. 9, Hot Blast, after first melting,	from				197,120	25,350		
Malleable Cast Iron,	about	48,000 Mr.						
IRON, WROUGHT:								
Good Bars, Bolts, Rivets, &c.,	average about	7	60,000	50,000	36,000		28,000,000	
Plates,	average about	480	60,000					
Different Sorts and Qualities—viz.,								
Bars, Rolled or Forged, mean,			57,555 K.					
Yorkshire,	from		66,390					
to	to		60,075 K.					
Yorkshire and Staffordshire Rivet,	from		59,740 Fa.					
Charcoal Bar,	to		63,620 Fa.					
Staffordshire,	from		62,231					
to	to		56,715 K.					
Lancashire,	from		64,795					
to	to		51,327 K.					
Lancashire,	from		60,110 K.					
to	to		53,775					
Swedish,	from		48,933 K.					
to	to		41,251					
Russian,	from		59,096 K.					
to	to		49,564					
Bushelled Iron from Turnings,	from		55,878 K.					
Hammered Scrap,	to		53,420 K.					
Angle-iron from various districts,	from		61,260 K.					
to	to		50,056					
Straps from various districts,	from		55,937 K.					
to	to		41,386					
Bessemer Cast Ingot,	from		41,242 W.					
Hammered and Rolled,	to		72,643 W.					
PLATES—mean—lengthwise,	from		50,737 K.					
crosswise,	to		46,171					
Yorkshire (Lowmoor), lengthwise,	from		64,200 Fa.					
crosswise,	to		62,490					
Yorkshire Bridge-iron, lengthwise,	from		49,930 Fa.					
crosswise,	to		43,940					
Yorkshire Plates, lengthwise,	from		58,487					
crosswise,	to		52,000 K.					
Staffordshire, lengthwise,	from		55,033					
crosswise,	to		46,221					
Staffordshire, lengthwise,	from		56,996					
crosswise,	to		46,404 K.					
Staffordshire Best-best Charcoal, lengthwise,	from		51,251					
crosswise,	to		44,760					
Staffordshire Best-best Charcoal, lengthwise,	from		45,010 Fa.					
crosswise,	to		41,420					
Best-best, lengthwise,	from		59,820					
crosswise,	to		49,945					
Best, lengthwise,	from		54,820 Fa.					
crosswise,	to		46,470					
Common, lengthwise,	from		61,280 Fa.					
crosswise,	to		53,820					
Bridge-iron, lengthwise,	from		50,820 Fa.					
crosswise,	to		52,825					
Lancashire, lengthwise,	from		47,600 Fa.					
crosswise,	to		44,385					
Lancashire, lengthwise,	from		48,865 Fa.					
crosswise,	to		45,915					
Lancashire, lengthwise,	from		53,849					
crosswise,	to		43,433 K.					
Durham, lengthwise,	from		48,848					
crosswise,	to		39,544					
PRECES CUT OUT OF LARGE FORINGS—								
Lengthwise,	from		47,582					
Crosswise,	to		43,759					
EFFECTS OF COLD ROLLING—								
Black Bar,	from		44,578 K.					
turned,	to		36,824					
cold rolled,	from		58,627					
Plate, cold rolled,	to		60,747 Fa.					
EFFECTS OF REHEATING AND ROLLING—								
Puddled Bar,	from		88,229					
five times piled, reheated, and rolled,	to		114,912					
eleven times,	from		43,904					
Loss of strength in screwed and chased bolts, from 7½ to 33½ per cent.—K.			61,824 CL.					
Loss of strength in welded joints, from 15 to 30 per cent.—K.			43,904					
HOOP—Best-best,								
Wire—Weak,			64,000 Mo.					
Average,			71,000 Mo.					
Strong Charcoal,			86,000 Ta.				25,300,000	
Very Strong,			100,000 G.					
Wire-ropes,			114,000 Mo.					
Lead Sheet,	11.400	711	90,000 Ra.				15,000,000 Ra.	
Platinum Wire, Hard,	21.000	151	3,328 Tr.				720,000 Tr.	
Annealed,			265,000 W.				24,240,000 W.	
			48,000				22,000,000 W.	
STEEL.								
Bars—Cast Steel, Rolled and Forged,	average	7.8	107,000 Mo.				29,000,000 Y.	
Blistered Steel, Rolled and Forged,	from		132,909 K.		269,000 F.			
Shear Steel,	to		92,015					
Bessemer Steel,	from		134,000 Re.				42,000,000 Mo.	
Rivet Steel,	to		104,208 K.					
Spring Steel,	from		118,468 K.					
Homogeneous Metal, Rolled and Forged,	to		111,460 K.					
Puddled Steel,	from		152,912 W.					
to	to		86,450 K.	63,796 K.				
Coleford Gun-Metal—weakest,	from		72,529 K.					
strongest,	to		90,647 K.					
mean of ten sorts,	from		89,724					
PLATES—								
Cast-steel—lengthwise,	from		93,000 Fa.					
crosswise,	to		89,724					
hard,	from		100,944 B (W.H).			57,500 B (W.H.)	23,883,000 B. (W.H.)	
soft,	to		71,484 K.					
Puddled Steel—lengthwise,	from		62,788					
crosswise,	to		90,000 Fa.					
Hard Steel, untempered,	from		94,782 Ma.					
tempered,	to		95,233 B (W.H)			52,500 } H (W. H.)	22,846,400 } B (W.H.)	
Soft Steel, untempered,	from		116,336 B (W.H)			63,750 } H (W. H.)	24,802,000 } B (W.H.)	
tempered,	to		108,970					
Increase in strength of steel by hardening in oil, from 12 per cent. to 70 per cent.—K.			160,540 Fa.					
Tin, Cast,								

TABLE OF PROPERTIES OF TIMBER.

COMMON NAME.	BOTANICAL NAME.	WHERE GROWN.	WEIGHT.		STRENGTH.					MODULUS OF ELASTICITY.
			Specific gravity.	Lb. in a cubic foot.	TENSILE FORCE, lb. on the square inch.	SHRINKING FORCE, lb. on the square inch.	CRUSHING FORCE, lb. on the square inch.	BREAKING FORCE, lb. on the square inch.		
				W.	F _t	F _s	F _c	F _b	F.	
Alder.	<i>Alnus glutinosa</i> .	Europe.	0.555 Tr.	34.6	14,186 M.	6,895 H.	9,540 Lb.	1,087,000 Tr.	
Apple.	<i>Pyrus Malus</i> .	"	0.793 M.	49.5	19,500 Bc.	6,399 H.	1,645,000 H.	
Ash.	<i>Fraxinus excelsior</i> .	Europe and Northern Asia.	0.753 Tr.	47.0	17,000 B.	9,000 H.	12,200 B.	1,525,500 Tr.	
Bambo.	<i>Bambusa arundinacea</i> .	Southern Asia.	0.400	25.0	14,130 Tr.	
Baracora.	<i>Erythrina Coriandrolendrum</i> .	Guiana.	0.807 Fo.	50.4	6,300 Bc.	8,818 Fo.	19,135 Fo.	579,000 Fo.	
Bartaball.	<i>Lucuma Bonplandii</i> .	"	0.640 Fo.	40.0	8,818 Fo.	19,048 Fo.	595,000 Fo.	
Bastard-box.	<i>Eucalyptus</i> (?)	Australia.	0.115 Fo.	69.6	9,790 Fo.	25,767 Fo.	1,636,000 Fo.	
Bech.	<i>Fagus sylvatica</i> .	Europe.	0.690 Eb.	43.1	11,600 B.	9,363 H.	9,363 H.	1,354,000 B.	
" West Indian.	"	West Indies.	0.843 Fo.	52.6	8,818 Fo.	20,331 Fo.	
Birch.	<i>Betula alba</i> .	Europe.	0.711 B.	44.4	15,900	6,402 H.	11,071 B.	1,615,000 B.	
Birch-American Black.	<i>Betula lenta</i> .	North America.	0.650 B.	40.6	11,663 H.	10,860 B.	1,477,000 B.	
Bitterwood.	<i>Sinaraiba excelsa</i> .	West Indies.	0.555 Fo.	34.6	5,511 Fo.	8,426 Fo.	
Blue-mahoe.	<i>Hibiscus tiliaceus</i> .	"	0.536 Fo.	33.4	8,818 Fo.	20,000 D.	
Blue-gum.	<i>Eucalyptus globulus</i> .	Tasmania.	0.917	57.25 D.	16,124 Fo.	800,000 Fo.	
Box.	<i>Buxus sempervirens</i> .	Australia.	0.843 Fo.	52.5	8,818 Fo.	20,331 Fo.	
Box of Ilawarra.	<i>Eucalyptus</i> .	Europe.	0.960 B.	59.9	20,000 B.	10,290 H.	
Broadleaf.	<i>Terminalia latifolia</i> .	Australia.	1.170 Fo.	73.0	9,921 Fo.	31,120 Fo.	1,421,000 Fo.	
Brown-bony.	<i>Terminalia latifolia</i> .	West Indies.	0.771 Fo.	48.1	7,716 Fo.	13,637 Fo.	635,000 Fo.	
Buckati.	"	Guiana.	1.034 Fo.	64.5	12,566 Fo.	27,276 Fo.	1,488,000 Fo.	
Bukuradda.	"	"	0.812 Fo.	50.7	9,921 Fo.	17,364 Fo.	
Bullet-tree.	<i>Achras Sideroxylon</i> .	"	1.029 B.	50.8	12,125 Fo.	21,323 Fo.	808,000 Fo.	
" Bastard.	"	West Indies.	1.046 Fo.	65.3	15,900 B.	2,020,000 B.	
Red.	<i>Bumelia salicifolia</i> .	"	0.902 Fo.	56.3	14,330 Fo.	22,315 Fo.	1,302,000 Fo.	
Cabacalli.	"	"	0.999 Fo.	62.3	11,023 Fo.	15,119 Fo.	992,000 Fo.	
"	"	Guiana.	0.893 Fo.	55.7	9,921 Fo.	12,394 Fo.	992,000 Fo.	
"	"	"	0.900 B.	56.2	9,921 Fo.	16,124 Fo.	850,000 Fo.	
Cabbage-bark.	<i>Andira inermis</i> .	West Indies.	0.945 Fo.	59.0	9,921 Fo.	15,119 Fo.	1,190,000 Fo.	
Cashash.	<i>Crescentia Cujato</i> .	West Indies-Tropical America.	0.557 Fo.	34.8	5,511 Fo.	10,164 Fo.	812,000 Fo.	
Cashaw.	<i>Prosopis juliflora</i> .	West Indies.	0.916 Fo.	57.2	9,921 Fo.	14,418 Fo.	744,000 Fo.	
Cedar of Lebanon.	<i>Cedrus Libani</i> .	Syria-Northern Africa-Europe.	0.480 Tr.	30.4	11,400 Bc.	5,660 H.	7,420 Tr.	486,000 Tr.	
" Red.	<i>Juniperus Virginiana</i> .	North America-West Indies.	0.650 Fo.	40.5	
" West Indian.	<i>Cedrola odorata</i> .	West Indies.	0.576 Fo.	36.0	6,614 Fo.	7,196 Fo.	
Chestnut.	<i>Castanea vesca</i> .	Europe.	0.535 Tr.	33.4	13,300 Ro.	10,556 Tr.	1,137,000 Tr.	
"	"	"	10,500 Bc.	
Cogwood.	<i>Laurus Chloroxylon</i> .	West Indies.	0.961 Fo.	60.0	12,122 Fo.	19,232 Fo.	890,000 Fo.	
Cowrie.	<i>Danmora Australis</i> .	New Zealand.	0.579 Tr.	36.2	10,960 H.	1,982,000 Tr.	
Crawwood.	<i>Xylocarpus carapa</i> .	Guiana.	0.603 Fo.	37.7	8,818 Fo.	12,394 Fo.	744,000 Fo.	
Cypress.	<i>Cupressus sempervirens</i> .	Europe.	0.655 Ro.	40.8	6,000 M.	
Dogwood-West Indian Black.	<i>Picidia Carthaginiensis</i> .	West Indies.	0.930 Fo.	58.0	11,023 Fo.	13,637 Fo.	812,000 Fo.	
Dogwood-West Indian White.	<i>Picidia Erythrina</i> .	"	0.943 Fo.	58.8	21,203 Fo.	
Ducallball.	"	Guiana.	0.910 Fo.	56.8	
Ebony-Oriental.	<i>Diospyros Ebenus</i> .	Eastern Tropical Islands.	1.136 Me.	71.0 Me.	13,228 Fo.	21,071 Fo.	1,692,600 Me.	
" West Indian.	<i>Brya Ebenus</i> .	West Indies.	1.193 Fo.	74.5	18,960 Fo.	27,276 Fo.	1,613,000 Fo.	
Eldor.	<i>Sambucus nigra</i> .	Europe.	0.895 M.	43.4	10,230	8,467 H.	
Elm-Common.	<i>Ulmus campestris</i> .	"	0.544 Eb.	34.0	13,489 M.	10,331 H.	6,078 B.	700,000 B.	
"	"	"	14,400 Bc.	9,720 Lb.	1,343,000 Tr.	
Fiddle-wood.	<i>Citharoxylon melano-cardium</i> .	West Indies.	0.707 Fo.	44.1	6,614 Fo.	12,394 Fo.	441,000 Fo.	
Fir-Red Pine.	<i>Pinus sylvestris</i> .	Northern Europe-Norway.	0.577 B.	36.1	14,300 Bc.	540 to 840 Bc.	5,575 H.	8,844 B.	1,458,000 B.	
"	"	"	0.480 Tr.	30.0	13,300 Bc.	9,540 Tr.	1,088,000 Tr.	
"	"	"	12,200 B.	6,200 H.	7,110 B.	
"	"	Prussia.	0.544 Tr.	34.0	9,540 Tr.	1,958,000 Tr.	
"	"	Scotland.	0.684 Tr.	42.7	7,323 Tr.	845,000 Tr.	
"	"	Norway.	0.512 Tr.	32.0	12,346 Tr.	1,804,000 Tr.	
"	"	"	0.698 B.	43.6	12,400 B.	9,864 B.	1,672,000 B.	
"	"	Britain.	0.555 Tr.	34.7	592 B.	8,370 Tr.	1,394,000 Tr.	
"	"	"	0.465 Tr.	29.1	10,296 Tr.	1,244,000 Tr.	
" American White Spruce.	<i>Abies alba</i> .	Northern America.	0.460	28.7	11,835 Tr.	1,633,000 Tr.	
" Weymouth Pine.	<i>Pinus Strobus</i> .	Northern America.	0.461 C.	28.8	1,600,000 Tr.	
" American Yellow Pine.	<i>Pinus variabilis</i> .	Northern America.	0.660	41.2	7,800 H.	5,445 H.	9,792 B.	1,226,000 B.	
" Pitch Pine.	<i>Abies resinosa</i> .	Northern America.	0.660	41.2	7,800 H.	9,792 B.	1,226,000 B.	
" Larch.	<i>Larix Europaea</i> .	Northern Europe.	0.496 Tr.	31.0	10,220 Ro.	970 to 1,700 Tr.	5,568 H.	4,992 to 6,894 B.	1,365,600 Tr.	

INDEX OF NAMES OF WOODS WHICH DO NOT OCCUR IN THEIR ALPHABETICAL PLACES IN THE TABLE

Acacia, Common or False, see Locust.	Gualacum,	see Lignumvitis.	Oak, Indian,	see Teak, Indian.	Quassia,	see Bitterwood.
Box, Australian,	Larch,	" Fir.	Pine,	" Fir.	Snakewood,	" Letterwood.
Deal,	Oak, African,	" Teak, African.	Plane, Common,	" Sycamore.	Spruce,	" Fir.

CHAPTER II.

OF THE STRENGTH OF A SHIP AS A WHOLE.

SECTION I.—PRINCIPAL STRAINING ACTIONS ON A SHIP.

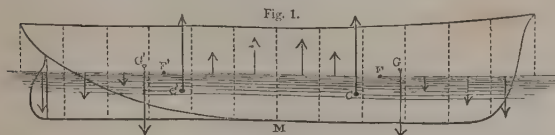
66. *General Explanations.—Gravity and Reaction.*—The straining actions to which a ship as a whole is subject, arise from differences between the distribution of the forces exerted from within, by the weight and reaction of the ship and her lading, and by the propelling machinery and sails, and the distribution of the forces exerted from without, by the water in supporting her and in resisting her motion. All those forces, taken as a whole, balance each other; and if they were so distributed, that to every force exerted from within, there were opposed an equal and opposite force exerted from without, and applied at the same point, there would be no straining action on the ship. The differences of distribution of the two systems of forces, produce straining actions which may be generally described as follows:—

A pair of equal and directly opposite forces, acting along the same line, but applied at different points, produces tension or compression according as the forces are directed from or towards each other:—

A pair of equal and opposite couples, exerted in the same plane, produces racking and bending:—

A pair of equal and opposite couples, exerted in parallel but different planes, produces twisting.

The *reaction* just mentioned is the force which the particles of the ship and her lading exert when they are compelled to accompany the movements of the waves. (See Division I., Articles 84 and 140.) From what is known of ocean waves, it may be inferred that the utmost effect of reaction is equivalent to an alternate increase and diminution of the weight of each particle to the extent of about *one-fourth part*;^{*} the increase being greatest in the lowest position of the ship, and the diminution greatest in her highest position.



67. *Longitudinal Racking and Bending.*—Let Fig. 1 represent a ship floating in smooth water, and let the vertical dotted lines represent transverse sectional planes dividing her length into any number of intervals, and her displacement into the same number of transverse layers.

It has already been explained, in Division I., Chapter III., Section I., Articles 86 to 99, that the whole upward pressure of the water, or buoyancy, is exactly equal to the whole weight of the ship; and that the centre of buoyancy is exactly in the same vertical line with the centre of gravity. If, moreover, the

weight of *each separate vertical layer* is proportional to its volume, and therefore equal to its buoyancy, in other words, if each layer is “*water-borne*,” the ship is free from longitudinal racking and bending when floating in smooth water.

But in general, some of the layers have an excess of weight over buoyancy, balanced by an excess of buoyancy over weight in other layers; and those excesses constitute a set of downward and upward vertical forces, which produce racking and bending actions on the ship, according to the principles explained in Article 44 of this Division, Case IV.

In sailing ships always, and often also in steamers, there is an excess of weight towards the head and stern, and an excess of buoyancy amidships; the vertical forces being such as are represented by the arrows in Fig. 1, and the effect, to make the ship arch upwards, or become what is called “*hog-backed*,” or simply “*hogged*.”

In steamers which have a long middle body, with the weight of the engines and boilers concentrated near its centre, there is often an excess of weight amidships as well as at the ends, and an excess of buoyancy between the middle and ends; the effect being to make the ship arch downwards, or “*sag*,” at the middle of her length, and “*hog*” near the ends.

In either of those cases, the process of finding racking forces and bending moments at a series of cross-sections of the ship is carried on as follows:—Compute the separate displacements of the several vertical layers, by the rules given in Division I. of this Treatise; then, supposing the weight of each layer to be known, compute the excesses of weight or of buoyancy in the several layers, and divide each by the thickness of the layer, so as to reduce them to excesses of weight or of buoyancy per foot of length; represent the quantities so computed by the ordinates of a curve, such as that shown in Fig. 8 of the preceding Chapter, and apply Rules M and N of Division III., Article 44. The results of those operations as regards the *greatest* racking forces and bending moment, in the case of a ship that tends to hog, are of the following kind:—

Let M be a transverse section which divides the ship into two parts, each separately water-borne (and which is seldom far from the midship section); then at the section M the racking force is nothing, and the bending moment is a maximum.

Let F and F' be a pair of cross-sections, at each of which the weight is just water-borne and no more; at these sections the racking force is a maximum.

Let G and G' be the centres of gravity, and C and C' the centres of buoyancy, of the two parts into which the section M divides the ship. (In the case represented by the figure, G and G' are further from M than C and C'.) Then the weight and buoyancy of the forward part, acting respectively through G and C, and the weight and buoyancy of the after part, acting respectively through G' and C', constitute a pair of opposite

^{*} Because the radius of the orbits of the surface-particles of water is not found to exceed one-fourth of the equivalent pendulum of the waves.

couples of equal moment; and that moment is the bending moment at the section M.

In the case of a ship that tends to sag amidships, the greatest bending moment is still at M, but the points G and G' are nearer to M than C and C'; so that the moment in question is contrary in direction to that represented in the figure, being a sagging moment instead of a hogging moment. There are also a pair of sections of greatest hogging moment, intermediate between M and the ends of the vessel; and there are four, instead of two, sections where the weight is exactly water-borne, and the racking force a maximum. Two of these lie between M and the sections of greatest hogging moment, and two between the sections of greatest hogging moment and the ends of the vessel.

The two following rules are evident:—

In ships of similar figures, with weights similarly distributed—

RULE I.—*The greatest racking forces are proportional to the displacements.*

RULE II.—*The greatest bending moments are proportional to the products of the displacements and lengths.*

The chief difficulty in applying those principles to practice arises from the complexity of the distribution of weights on board. If we make the assumption, that one-half of the weight is exactly water-borne, or distributed proportionally to the displacement of each transverse layer, and the other half distributed uniformly over the length of the ship, we arrive at results of which the following are examples:—

TABLE A.

FIGURE OF VESSEL.	In smooth water, greatest	
	Racking Force = Displacement ×	Hogging Moment = Dispt. × Length ×
	0	0
CASE I.—Water-lines and cross-sections rectangular,	0	0
" II.—Water-lines harmonic curves; cross-sections rectangular, ...	0.053	0.017
" III.— <div style="display: inline-block; vertical-align: middle;">Water-lines harmonic curves; cross-sections triangular, ... Also, Water-lines wave-shaped; cross-sections rectangular, ...</div>	0.08	0.025

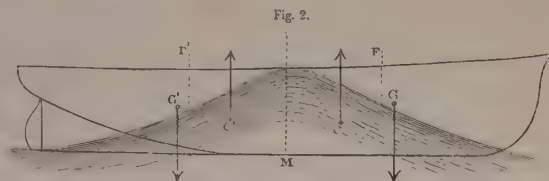
If the ship floats *amongst waves*, two extreme cases may be distinguished, between which all the others will be found; viz., the case of a vessel so small compared with the dimensions of the waves, that she simply accompanies the motion of the surface-particles of water; and the case of a vessel so large, that she stretches over at least the entire length of a wave, and that the heaving motion of her centre of gravity is comparatively small.

In the former case it is sufficient to take into account the straining effect of reaction, in addition to that of gravity; that is to say, to add *one-fourth part* to such numbers as those given in Table A.

In the latter case, little or nothing is to be added for reaction; but the straining effects of the unequal distribution of buoyancy must be taken into account; and these are far more severe than any that can be produced by reaction. In fact, the most severe straining action of the waves on a ship takes place when her centre of gravity remains at a constant level, so that there is no reaction; and this, therefore, is the case to be considered in determining the strength that a given ship requires.

Fig. 2 represents a ship in the position now referred to, being supported amidships on the crest of a wave of such dimensions,

that her head and stern are left wholly unsupported in the centres of two successive wave-troughs.



To find the straining actions produced by the wave, in this case—measure upon each cross-section of the body-plan of the ship, the difference between the area immersed in smooth water, and the area immersed in the wave: supposing those differences to be measured in square feet, reduce them to tons per foot of length by dividing by 35; these quantities will represent excesses or deficiencies of buoyancy per foot of length at each cross-section, according as the depth of immersion in the wave is greater or less than in smooth water; take these excesses and deficiencies as the ordinates of a *curve of loads*, and proceed by Rules M and N of Article 44 of this Division, Case IV., to find the racking forces in tons, and the bending moments in foot-tons.

With the racking forces and bending moments thus found, are to be combined the racking forces and bending moments due to differences between the distribution of weight and of buoyancy in smooth water. The results will be the total racking forces and bending moments; and will be of the following kind:—Let M be a cross-section which divides the ship and the wave into two parts, such that each part of the ship is separately water-borne by the part of the wave which it displaces. That cross-section will always nearly, and sometimes exactly, coincide with the midship section of the ship, and the crest of the wave, and will be the place of greatest bending moment, tending to make the ship hog.

Let G and G' be the centres of gravity of the two parts of the ship, and C and C' the centres of the two parts of the wave which they respectively displace. The weight and buoyancy of the forward part, acting respectively through G and C, and the weight and buoyancy of the after part, acting respectively through G' and C', constitute a pair of opposite couples of equal moment; and that moment is the bending moment at M.

The greatest racking forces occur, as before, at the two sections, F and F', where the weight is just water-borne and no more.

The following are examples of the results, for vessels of the forms already described in Table A, half the weight being supposed water-borne in smooth water, and the other half uniformly distributed along the vessel's length:—

TABLE B.

CASE	Greatest	
	Racking force = Displacement ×	Hogging Moment = Dispt. × Length ×
	0.00	0.00
I.—In smooth water, as in Table A,	0.00	0.00
Effect of wave,	0.16	0.05
Combined effect,	0.16	0.05
" II.—In smooth water, as in Table A,	0.053	0.017
Effect of wave,	0.096	0.027
Combined effect, nearly,	0.149	0.044
" III.—In smooth water, as in Table A,	0.080	0.025
Effect of wave,	0.069	0.017
Combined effect, nearly,	0.149	0.042

It thus appears that while the straining actions in smooth water become greater as the figure of the vessel becomes finer and sharper, the additional straining actions produced by waves become less, and that those two opposite changes in a rough way compensate for each other; and also that the following rules for calculating approximately the greatest racking force and hogging moment give limits which those quantities are not likely to exceed in any case of ordinary occurrence in practice:—

RULE III.—For the greatest racking force, take $\frac{1}{100}$ of the displacement;

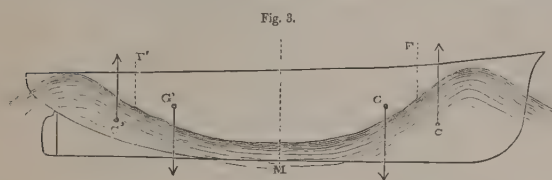
RULE IV.—For the greatest hogging moment, multiply the displacement by $\frac{1}{25}$ of the length.

For example, in a ship of 2680 tons displacement, and 205 feet long, those rules give—

For the racking force, $2680 \times 0.16 = 429$ tons;

For the hogging moment, $\frac{2680 \times 205}{20} = 27470$ foot-tons.

Fig. 3 represents the condition of a ship, having an excess of



buoyancy at the ends, produced by the crests of two waves, and a deficiency of buoyancy in the middle, produced by the intervening trough; the result being that the action of the waves tends to make the ship sag.

But it is unnecessary to enter into detailed calculations respecting this position of the ship; because, in all cases of ordinary occurrence in practice, the sagging moment thus produced is less severe than the hogging moment produced when the ship is balanced on the crest of a wave; and the strength which is sufficient to resist the hogging action is sufficient to resist the sagging action.

Up to this point the longitudinal racking and bending actions upon a ship have been treated as if they took place in her longitudinal midship plane. But through the rolling of the ship and of the waves, those actions very often take place in oblique longitudinal planes, making great angles with the longitudinal midship plane; and that fact must not be lost sight of in distributing the materials of the ship, as will be more fully explained in Section II. of this Chapter.

68. *Transverse Bending*.—The most severe case of that sort of action which may be called transverse or thwartships bending is that which takes place when the ship takes the ground, and her whole weight rests on her keel. There is then a tendency to split the ship's bottom lengthwise, through the breaking across of the floors, or lowest parts of the frames; and the bending moment which expresses that tendency is found as follows:—Let Fig. 4 represent a cross-section of the ship; D, D, the weights of the two halves into which the ship is divided by her longitudinal midship plane, each half being regarded as concentrated at its own centre of gravity; C, the upward pressure of the ground upon the keel, being equal and opposite to the whole weight or displacement of the ship; then the bending moment in question is equal to *half the displacement multiplied by the horizontal distance of D from C*.

It is difficult to make any precise calculation of that distance;

but it may be roughly estimated in most cases as not differing greatly from *one-fourth of the mean breadth of the plane of flotation*: so that the transverse bending moment may be estimated approximately at the following value:—

$$\frac{\text{Displacement} \times \text{Extreme breadth} \times \text{Co-eff. of fineness of L.W.L.}}{8}$$

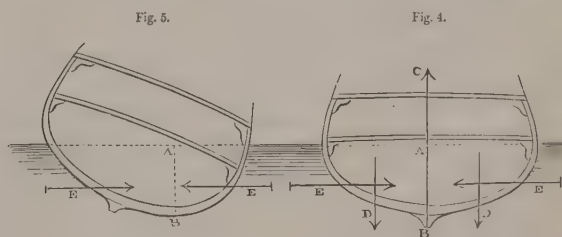
The transverse bending moment of any particular transverse layer of the ship may be estimated approximately on the same principle, as having the value—

$$\frac{\text{Weight of the layer} \times \text{its breadth}}{8}$$

The use of stanchions or pillars to support the deck beams has a tendency to diminish the transverse bending action.

When a ship is afloat, a transverse bending moment may act in the contrary direction to that which acts when she takes the ground, owing to the concentration of the weight and load of the masts directly over the keel; but this action cannot be in any case so severe as that which has already been considered.

69. *Transverse Compression*.—The materials with which the vessel is laden, being for the most part solid, do not exert any appreciable outward transverse pressure. It follows that the equal and opposite transverse components of the pressure exerted from without by the displaced water (which may be represented by the arrows, E, E, in Figs. 4 and 5), having no pressure from within to



resist them directly, produce a straining action, tending to make the ship collapse or flatten transversely. The kind of stress which this action produces depends on the position of the vessel. When she is upright, as in Fig. 4, it consists mainly of a thrust along the beams and floors, and of an inward bending action upon the frames at the bilges; when she heels over at a great angle, as in Fig. 5, it consists mainly of a transverse racking action, straining at once the bilges, and the knees which connect the beams with the frames. In a ship with transverse partitions or bulkheads, these are compressed directly or obliquely athwartships, as the case may be.

The amount and position of the resultant of this kind of straining pressure are thus determined:—

RULE I.—If the draught of water is uniform: *In any position of the ship, upright (as in Fig. 4), or inclined (as in Fig. 5), multiply half the square of the draught of water (AB), by the length; the weight of the volume of water thus found is the amount of the required resultant: that is, if the dimensions are taken in feet, divide the product by 35 for tons, or multiply by 64 for lbs. The position of the resultant is at two-thirds of the draught below the surface of the water.*

The same rule applied to any particular cross-section of the ship gives the position of the resultant, and its amount per foot of length at that cross-section.

RULE II.—If the draught of water is not uniform:—*Take the*

length of the ship as the base of a curve, and the half-squares of the draughts of water as ordinates; find the area of the curve by Simpson's Rule: it will represent a volume of water whose weight will be the amount of the resultant required. Then take the third parts of the cubes of the draughts of water for the ordinates of a new curve; find its area by Simpson's Rule, and divide it by the area of the former curve; the quotient will be the depth of the resultant below water.

The most severe lateral compression on the sharp ends of all ships, and on the middle also, in narrow, deep, and sharp-floored vessels, occurs when the ship is upright. But in comparatively broad, shallow, and flat-floored vessels, the lateral compression on the middle often becomes more severe when the vessel is heeled over to a great angle relatively to the surface of the water, because of the depth of immersion being increased. The greatest value of that angle may be roughly estimated by supposing the leeside to be immersed to the gunwale.

In applying the foregoing principles to the upright position of the ship, the draughts of water are to be taken exclusive of the keel and deadwood; because the equal pressures exerted on the opposite sides of those parts of the frame have no straining effect on the ship as a whole. A distinction is also to be drawn, in those ships which have deep floors, between the transverse pressure transmitted directly through the floors, and tending simply to compress those pieces of the frame, and that which acts on the bottom and sides above the level of the floors, and tends to make the bilges collapse. The following is the rule for separating those parts from each other:—

RULE III.—Measure the depth of immersion to the upper edge of the floors, and use Rule I. or Rule II. to compute the pressure tending to make the bilges collapse. Then measure the depth of immersion to the upper side of the keel, and use Rule I. or Rule II. to compute the whole pressure: the difference will be the pressure transmitted directly through the floors.

For example, suppose that in an iron ship we have the following data:—

Depth of immersion to upper edge of keel,	Feet.
" of floor-plates,	19½
" of immersion to upper edge of floor-plates,	2
	17½

Then—

Transverse pressure per foot of length, above level of floors, =	Tons.
$\frac{(17\frac{1}{2})^2}{2 \times 35}$	= 4.375
Total transverse pressure per foot of length,	$\frac{(19\frac{1}{2})^2}{2 \times 35}$ = 5.432
Difference, transmitted directly through floor-plates,	1.057

When the ship is in an inclined position, the same rule may be applied to distinguish between the transverse pressure transmitted directly through the ribs, and that which tends to rack the framework of the vessel.

For example, the same ship from which the preceding measurements were taken, being heeled over till the lee gunwale is at the surface of the water, measures at the midship frame 21 feet perpendicularly from the plane of flotation to the turn of the bilge outside the skin, and 20½ feet to the inside edge of the frames; therefore, in that inclined position, the transverse pressure per foot of length at the midship frame is thus made up^o—

Above inside edge of frames,	$\frac{(20\frac{1}{2})^2}{2 \times 35}$	Tons.
		= 6.004
Total per foot of length,	$\frac{(21)^2}{2 \times 3}$	= 6.300
Difference, transmitted directly through ribs,		0.296

It may sometimes be necessary to find the resultant of the transverse pressure on some portion of the bottom of the vessel which is not bounded by vertical cross-sections; and then the rule is as follows:—

RULE IV.—Measure the area of the part in question, and find the centre of its projection on the sheer plan; the amount of the pressure will be equal to the weight of a volume of water found by multiplying that area by the depth of that centre below the plane of flotation.

The position of the resultant of that pressure may be found by determining the centre of pressure of the projected area just mentioned, by the method of Division I., Article 63, regard being had to the fact that the intensity of the pressure at any point is proportional to the depth below the plane of flotation. When the projected area is a long narrow parallelogram, divide the difference of the cubes of the depths of immersion of its ends by the difference of their squares; two-thirds of the quotient will be the depth of immersion of the centre of pressure.

70. *Straining by the Propelling Apparatus.*—The propelling instrument, whether consisting of a pair of paddle-wheels, a screw, or any other contrivance, exerts a forward pressure upon the vessel equal and opposite to the resistance of the water, which may be computed for a given speed by the method of Division I., Article 163.

Another way of computing that pressure is to divide the effective or net power of the engine, reduced to foot-pounds per minute, by the speed of the vessel in feet per minute; the quotient will be the pressure in pounds; or to multiply the indicated horse-power of the engine by 200, and divide by the speed in knots.

The kind of straining action produced by that pressure depends on the place where it is applied. In the case of a screw, it acts nearly on a level with the resultant resistance of the vessel; so that its action on the ship as a whole is to produce longitudinal compression of the part below water. In the case of a pair of paddle-wheels, it is applied to the bearings of the paddle-shafts, at a certain height above the resultant resistance; and it forms, with that resistance, a couple which is balanced by an equal and opposite couple of vertical forces, consisting of an increase of the upward pressure of the water on the forward part of the vessel, and a diminution of that pressure on the after part. That combination of couples tends to produce longitudinal racking of the vessel in one direction throughout its whole length, and at the same time a bending moment in a longitudinal vertical plane, which is most severe close to the vertical plane of the paddle-shaft, and which is approximately equal to the forward pressure multiplied by one-half of the height of the paddle-shaft above the resultant resistance.

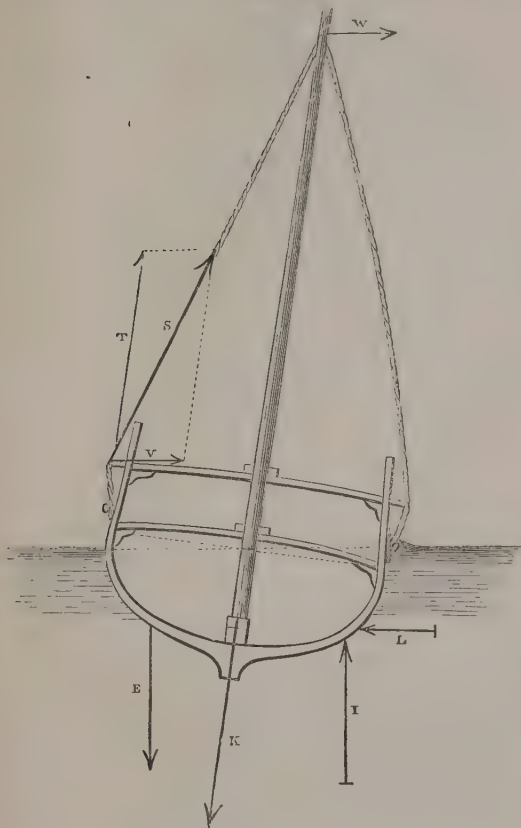
The directions of the straining actions of the propelling apparatus are reversed when the ship is driven stern foremost.

As regards the ship as a whole, the straining actions of the propelling apparatus are in most cases comparatively trifling. They are not so, however, as regards the particular pieces of the framework to which they are directly applied; and they will therefore be further considered in the Fourth and Sixth Divisions.

^o The shape of the ship here used as an example resembles that of the *Formby* (Plates $\frac{F}{1}$, $\frac{F}{2}$), but is two feet narrower and draws a foot more water.

71. *Straining by Sails.*—The longitudinal straining action of sails is like that of paddle-wheels. The forward pressure of the wind on the sails, whose resultant acts through the centre of effort (see Division I., Article 181), and the equal and opposite longitudinal component of the resistance of the water, form a couple tending to depress the bow and raise the stern, which is resisted by an equal and opposite couple exerted by the vertical

Fig. 6.



pressure of the water; and that pair of couples tends to rack the whole ship in a longitudinal vertical plane: but the practically important straining effects of this action are not those which regard the ship as a whole, but those which regard the particular parts to which the forces are applied.

The transverse straining action of sails takes place in the following manner:—

W represents the transverse component of the pressure of the wind, acting through the centre of effort of the sails; L, the equal and opposite lateral resistance of the water, produced by the leeway of the ship, and acting through the centre of lateral resistance. This couple of forces heels the ship over, until it is balanced by the equal and opposite couple of pressures, I and E; I acting upwards on the immersed half of the ship, and E acting downwards on the emerged half. The moment of the couple formed by I and E, is the moment of stability corresponding to the angle of heel produced by the pressure of the wind on the sails.

The lateral pressure of the wind, W, is transmitted to the vessel in the following manner:—K represents a thrust, exerted by the masts upon the keelson. S represents the resultant tension of the weather shrouds and backstays, exerted on the weather channels,

and resolved into two components—a horizontal component V, parallel and equal to W, and opposite to L; and a component T, parallel, equal, and opposite to the thrust of the mast, K. The moments of the two couples (V, L), and (T, K), are together equal and opposite to, and are balanced by, the moment of the righting couple (E, I).

The greatest bending moment produced by these forces is obviously that which is exerted upon the floors immediately to leeward of the keel, tending to compress their inner edges and stretch their outer edges; being the moment of the force, I, relatively to the midship plane of the vessel; and for practical purposes it may be estimated as being approximately equal to *one-half of the righting moment corresponding to the greatest angle of steady heel produced by the sails alone*; that is to say—

Bending moment on floors = Displacement \times height of metacentre above centre of gravity \times sine of angle of steady heel $\div 2$.

For example, suppose—

Height of metacentre,..... 4 feet;

Angle of heel $14\frac{1}{2}^\circ$; sine of angle of heel, $\frac{1}{4}$;

Then—

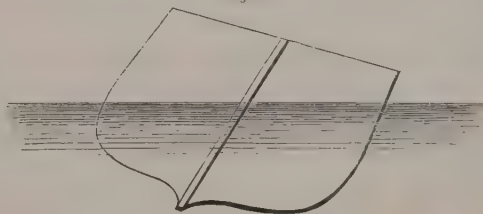
Bending moment = Displacement $\times \frac{1}{2}$ foot of leverage.

72. *Straining by Reaction in Rolling.*—When a ship performs rolling oscillations, each particle exerts a reactive force proportional to and nearly in the direction of its displacement from its position of equilibrium.

Those reactions have a moment, which is balanced by the equal and opposite moment of the righting couple exerted by the water. But the righting couple consists wholly of vertical forces; whereas the reactions consist of horizontal and vertical forces combined.

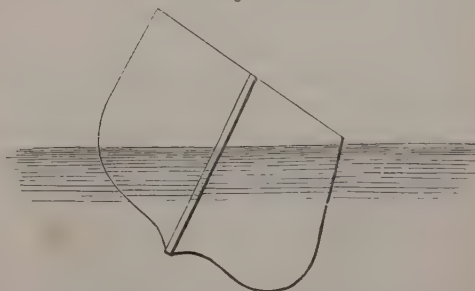
The result is, that when a ship performs rolling oscillations to one side and to the other of a steady position, whether upright or inclined, there is a *racking moment* exerted, tending to distort her

Fig. 7.



in the manner shown in Fig. 7; and that moment is equal to the moment of the horizontal components of the reactions of her particles. It may be calculated approximately as follows:—Find,

Fig. 8.



from the stability of the ship, the *righting moment* corresponding to her greatest angle of deviation from her steady position; multiply

it by the square of her depth, and divide by the sum of the squares of her breadth and depth.

It is to be observed, that the direction of the distortion thus produced by reaction, is contrary to that of the distortion which is produced by the transverse compressing action of the water, and which is represented in Fig. 8; so that those two straining actions cannot co-operate, and may to a certain extent counteract each other.

73. *Twisting* may take place when a ship is exposed at different parts of her length to forces tending to heel her in opposite directions: for example, when she lies obliquely to the waves, and her head and stern float upon opposite slopes; but it is unnecessary to consider this action in detail, for the stress which it produces must be less severe than that due to longitudinal bending.

SECTION II.—RESISTANCE OF A SHIP TO THE PRINCIPAL STRAINING ACTIONS.

74. *General Structure of a Ship.*—The one essential part of a ship's structure, without which she cannot exist, is the *skin*. It is possible to build a ship which shall be nothing but a skin, without framework; and some ships have been built, and successfully used, which very nearly answer that description.

As regards both the exclusion of water and the resistance to straining actions, the *planking of the decks* may be regarded as part of the skin. When there is an *inner skin*, it is strained in the same manner, though not to the same extent, with the outer skin.

The materials of the skin (as already stated in Division II., Article 31) are usually arranged in *strakes*, or longitudinal bands: the *seams*, or longitudinal joints of the strakes, follow nearly the course, below water, of normal lines (Division II., Article 26), and above water, of sheer-lines (Division II., Article 27); while the butts, or transverse joints, are parallel or nearly parallel to the frames or transverse ribs. There are exceptions, however, to this arrangement, where the skin consists of diagonal strakes.

In an iron vessel, the plates of the skin are rivetted at the seams to each other, and at the butts to *covering-straps*, by means of which they are connected together; and thus the whole skin is made into one continuous shell, capable of exerting tensile stress at any point and in any direction. In a wooden vessel, tensile stress can only be transmitted from one plank to another by the aid of the frames to which the planks are fastened, or by making the skin of two or more layers of planks running in different directions, and fastened together (for an example of this, see the Plates of the *Victoria and Albert*, especially Plates $\frac{9}{2}$, $\frac{9}{3}$). There is an exception to this in the case of clinker-built boats, in which the lower edge of each strake of plank overlaps, and is nailed or pinned to, the next strake below.

The uses of the pieces of which the *framework* of a ship consists may be classed under three heads, as follows:—

I. To connect the pieces of the skin together, so as to transmit tension;

II. To stiffen the skin against buckling by compression;

III. To add their strength to that of the skin, and so to render a thinner skin sufficient than would otherwise be necessary;

IV. To distribute loads that would otherwise be too much concentrated.

According to the most ancient and ordinary structure of the framework of a ship, its principal pieces may be thus classed with

reference to the manner in which they contribute to the strength of the ship:—

I. Transverse pieces which connect the strakes of the skin together, and resist transverse compression and transverse racking and bending—that is to say, the *ribs* or *frames*, whether square or cant (Division II., Articles 23, 28), and the *breast-hooks* and *transoms* (Division II., Article 29). With these may also be classed the *deck-beams*, which connect together the planks of the deck, and support them and their load, and also help the frames (with which they are connected by *knees*) to resist the various transverse straining actions; and the *bulk-heads* or transverse partitions, by which ships, especially those built of iron, are divided into compartments.

II. Longitudinal pieces which help the skin to resist longitudinal bending actions; comprising the *keel* and *keelsons*, respectively below and above the ship's floor; *shelf-pieces* and *water-ways*, running round below and above the ends of the deck-beams; *carlings* and *stringers*, running along the decks; the *longitudinal frames* sometimes used in iron ships, which follow nearly the course of normal lines, &c. The keelsons also serve to distribute loads, such as the thrust of masts and the weight of engines, which would otherwise be too much concentrated.

III. The *stem* and *stern-post*, which are upward continuations of the keel, and contribute to the strength of the ship as a whole chiefly by protecting the ends of the strakes of the skin against blows; but which have also special duties to perform; the stem by partly carrying the bowsprit, and the stern-post by supporting the rudder, and in some cases the screw.

IV. *Diagonal braces*, to assist the skin, decks, and bulkheads, in resisting racking forces.

V. Pieces for bearing direct vertical loads; such as the *pillars* or *stanchions*, which transmit the load of the decks down to the main or middle keelson. All the upright or nearly upright pieces previously mentioned, such as the stem, stern-post, ribs of the sides, &c., have to bear more or less direct vertical load; but its straining effect on these is inconsiderable compared with that of the other actions to which they are subjected.

The preceding classification comprehends only the main or leading parts of the framework of a ship, and makes no mention of many subordinate parts, which will be fully described in the sequel; neither does it include various contrivances used in special modes of shipbuilding, which will also be explained further on.

The dimensions necessary in order to give sufficient strength to the various parts of a ship, are in ordinary practice determined by rules, which have been deduced from an immense number of practical trials, in the course of the long period during which shipbuilding has been cultivated as an art. Such rules form useful data in applying scientific principles to the strength of ships; and they will be frequently referred to in the sequel.

75. *Total and Effective Sections—Factors of Safety—Working Moduli of Strength—Combined Materials.*—The distinction between total and effective sectional areas has been explained in Article 20 of this Division.

In an *iron ship*, that distinction has to be considered chiefly at the *butts* of the plating. These are most usually double-rivetted to a covering-strap; and therefore, agreeably to what has been stated in Article 30 of the present Division, a deduction of about *three-tenths* may be made from their sectional area for rivet-holes; but as the butts may be made to *shift* or break joint in such a way

that in a given cross-section of the ship there is only one butt to three or four strakes of plating, that deduction amounts to only *one-tenth* or thereabouts of the entire area of plates traversed by the given cross-section; and further, that deduction may be regarded as neutralized by the additional area due to the overlaps at the seams of the plating; so that if *the girth of a given cross-section be multiplied by the mean thickness of the plating*, the product may in most cases be regarded as practically equal to the *effective sectional area*.

The *seams* of the plating have to transmit the racking stress, due to the racking force in a longitudinal plane. According to a mode of building which was introduced simultaneously and independently by Mr. Scott Russell and Mr. J. R. Napier, and is now universally practised, the strakes of plating are put on in an inner and outer layer; so that each strake of the outer layer overlaps at its upper and lower edge the two adjoining strakes of the inner layer. Between each outside strake and each frame that it crosses, is inserted a "filling-piece" of a thickness equal to that of the inside strakes, and of a breadth equal to that of the frame. The seams are in some vessels double-rivetted, in others partly double and partly single, and in others single-rivetted; and agreeably to what has been stated in Article 30 of this Division, the effective area may be computed by multiplying the total area by the following factors:—

	Effective area =
	Total area ×
For double-rivetted seams,.....	0.70
For single-rivetted seams,.....	0.56

In a *wooden ship*, the "shift" of plank is so arranged that in any given cross-section there are at least four strakes of plank to each butt; so that a deduction of *one-fourth* has to be made from the total sectional area on account of butts. Deducting about one-ninth of the remaining three-fourths for treenail-holes or bolt-holes, it appears that, in a wooden ship, *the effective sectional area of the skin for resisting tension may be taken at two-thirds of the total sectional area*. One convenient way of making this allowance, is by increasing the factor of safety in the proportion of 3 to 2; for example, if 10 be the proper factor of safety for a solid piece of timber, 15 will be the factor for the skin of a ship under tension.

With respect to *factors of safety* in shipbuilding, it is to be remarked, that amongst the straining forces to which a ship is subjected as a whole, no distinction can be drawn between dead load and live load, because the whole load is in motion; and therefore that one factor of safety serves for the whole; so that calculations may be made with a "working modulus of strength," as explained in Article 46 of this Division. At the same time, the motions of the load are never so abrupt and violent as those of the live load on land structures, such as railway viaducts; and therefore a factor of safety intermediate between those suited for dead and live loads upon land structures is sufficient. As will afterwards be shown by examples, very various factors of safety and working moduli of strength occur in actual ships; but as regards the most severe and important straining actions upon the skin, the following may be taken as average results of good practice:—

	Modulus of Rupture about		Factor of Safety for total sectional area.	Working Modulus of Strength.	
	Tons on the sq. inch.	Lbs. on the sq. inch.		Tons on the sq. inch.	Lbs. on the sq. inch.
Iron skin, under tension,.....	20	= 44,800	5	4	= 8,960
" under compression,.....	12	= 26,880	5	2.4	= 5,376
Wooden skin, under tension,.....	5.625	= 12,600	15	0.375	= 840
" under compression, 8		= 6,720	10	0.3	= 672

The factor of safety and working modulus are applicable in each case to the *total sectional area*.

Steel may be taken as being from 1.6 times to double the strength of wrought iron; but no working moduli for steel are inserted in the preceding Table; because practical experience has scarcely yet furnished sufficient data to enable such moduli to be stated with perfect confidence.

The sum of the two working moduli for timber is very nearly *one-tenth* of the corresponding sum for iron; and hence two ships of equal size and similar figure, built respectively of wood and iron, are equally strong when the transverse sectional area of material in the wooden ship is *ten times* the corresponding area in the iron ship.

When wood and iron are so arranged as to act side by side in resisting a load—for example, when the wooden planks of a deck co-operate with iron stringers and waterways, and an iron skin, in resisting tension—the wood undergoes the same elongation with the iron, and therefore exerts a stress per square inch smaller than that exerted by the iron, in the proportion in which, not its modulus of strength, but its *modulus of elasticity* is smaller; that is to say, in the proportion of about 1 to 16. Therefore, when wood and iron co-operate in this way, the sectional area of wood may be reduced to an *equivalent area of iron*, by *dividing by 16*.

76. Resistance to Longitudinal Bending.—The ship resists the longitudinal bending moment described in Article 68, in the manner of a tubular girder; the skin, decks, keel, stringers, and longitudinal pieces generally at each cross-section, exerting tension or thrust according to their position relatively to the neutral axis of the cross-section, and so producing a sufficient moment of resistance. The best way of insuring that the moment of resistance of the ship shall be sufficient in all positions into which she can roll, is to arrange the material so as to make her moment of resistance to bending in a *horizontal* plane at least as great as her moment of resistance in a *vertical* plane. That condition is almost always fulfilled by ships which are not very narrow in proportion to their depth.

The **FIRST STEP** in calculations respecting the working moment of resistance of a given ship is to find the *neutral axis of the strained parts of the midship section*, which is done by the application of Rule I. of Article 46 of this Division; the wooden decks of iron ships being reckoned as equivalent to $\frac{1}{16}$ of their area of iron.

As the most severe longitudinal bending action, being that which takes place when the ship is balanced upright on the crest of a wave (as in Article 68, Fig. 2, of this Division), brings the upper part of the ship into tension, and the lower part into compression, the best position for the horizontal neutral axis of the midship section is such as to divide the depth from gunwale to keel in the proportion fixed by the principles explained in Article 48 of this Division; that is to say, in iron ships, the neutral axis should be at about *five-eighths* of the depth below the gunwale, and *three-eighths* of the depth above the keel; and such is found to be actually the case in good examples of iron ships. In wooden ships, the best proportion is less definitely known; but the actual proportion in good examples is very nearly the same as in iron ships.

The neutral axis for a horizontal bending action is the upright axis at the middle of the breadth.

The **SECOND STEP** is to find the *moment of inertia of the strained parts of the midship section*, as explained in Rule II. of Article 46, and *divide it by the distance of the most severely strained part from the neutral axis*; the quotient is the *moment of resistance with the working modulus unity*.

In computing the moments of inertia, areas are most conveniently expressed in inches; but depths may be expressed in feet, in order to avoid the large numbers produced by expressing them in inches, provided moments are expressed in foot-tons or foot-pounds, and not in inch-pounds.

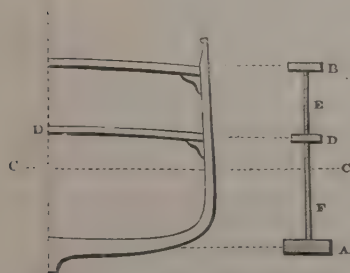
The **THIRD STEP** may be one or other of the two following processes:—To find the moment of resistance of a ship with a given working modulus of strength, multiply the result of the second step by that modulus; or, to find the actual modulus or greatest stress in a given ship under a given bending moment, divide that moment by the result of the second step.

Although the most severe bending action is one which tends to stretch the upper and compress the lower parts of the ship, it must be borne in mind, that she is very often subjected to a bending action in the reverse direction, as represented in Fig. 3 of this Chapter; and therefore that all her longitudinal pieces must be so arranged and fastened as to be capable of bearing either thrust or tension.

Although the cross-section for purposes of resistance to bending is in general held to terminate at the gunwale, or at the level of the uppermost complete deck, the rails and bulwarks which rise above that level being excluded from the calculation, yet there are some cases in which a bridge-deck, or platform, is so framed as to form the upper boundary of the effective section (see, for example, the midship section of the *Persia*, in Plate $\frac{1}{2}$, where some strength is gained by means of iron stringer-plates in the platform, about 7 feet above the upper deck); and there are other cases in which the bulwarks and rail are so strongly framed, that the rail may be regarded as the upper boundary of the effective section.

The two following examples are taken from ships described by Mr. John Vernon, in a paper read to the Institution of Mechanical Engineers, in August, 1863. One ship is of iron, the other

Fig. 9.



of wood; each is of 1200 tons register, and 2680 tons displacement at 20 feet draught of water. Their figures are exactly similar, the midship sections being nearly rectangular, and sharp at the bilge; so that for the purpose of the calculation of resistance to longitudinal bending, the ship may be treated as a sort of I-shaped girder (see Fig. 9): the keel, keelsons, and skin of the bottom being regarded as if collected in a flange, A; the sides, with their stringers, in a web, EF; the upper deck, with its stringers and water-ways, and the gunwale and part of the sheer strakes, in a flange, B; and the lower deck, with its stringers and shelf-pieces, in a flange, D; and the depths being as follows:—

$$BD \ 7\frac{1}{2} \text{ feet} + DA \ 16 \text{ feet} = 23\frac{1}{2} \text{ feet.}$$

The neutral axis is represented by CC.

EXAMPLE I.—IRON SHIP.

	Sectional Area.	Sq. inches.
D, Iron,.....	113
Wood $960 \div 16 =$ equivt. iron,.....	60
E,	173
B, Iron,.....	120
Wood $804 \div 16 =$ equivt. iron,.....	55
F,	50
A,	105
.....	320
.....	493
Total,.....	1211

Of this area 797 square inches, or very nearly *two-thirds*, are skin; the remainder consists of longitudinal pieces of the frame (such as keel, keelsons, stringers, &c.), and of wooden decks reduced to equivalent areas of iron.

STEP I.—Determination of the neutral axis:—

	Area. Sq. inches.	Depths below B. Feet.	Products.
B,	173	0	0
E,	120	3.75	450.0
D,	105	7.5	787.5
F,	320	15.5	4960.0
A,	493	23.5	11585.5
1211 total area,.....	17783.0 sum.

$$\text{Neutral axis below B,.....} \quad 14.7 \text{ feet} = \overline{BC}.$$

Proportions in which the neutral axis divides the depth:—

$$\begin{array}{rcl} \overline{AB} & : & \overline{BC} : \overline{CA} \\ : & : & 23.5 : 14.7 : 8.8 \\ : & : & 8 : 5 : 3 \text{ very nearly.} \end{array}$$

STEP II.—Calculation of moment of inertia, and of moment of resistance to the modulus unity:—

	Area. Sq. inches.	Multipliers.	Products.
B,	173	$(14.7)^2$	87383.57
E,	120	$(10.95)^2$	14388.30
D,	120	$(7.5)^2 \div 12$	562.50
D,	105	$(7.2)^2$	5443.20
F,	320	$(0.8)^2$	204.80
F,	320	$(16)^2 \div 12$	6826.67
A,	493	$(8.8)^2$	38177.92

$$\text{Divide by } \overline{BC} = 14.7 \quad \dots\dots\dots) \ 102986.96 \text{ sum.}$$

$$\text{Moment of resistance to the modulus 1,.....} \quad 7006 \text{ nearly.}$$

STEP III.—Calculation of the greatest stress, or actual modulus, with a known bending moment:—

Greatest Bending Moment, as computed in Article 68.	
Divide by.....	7006) 27470 foot-tons.
GREATEST TENSION,.....	3.92 tons on the square inch;

And inasmuch as $\overline{CA} = \frac{2}{3} \overline{BC}$, we have—

$$\text{GREATEST THRUST} = \frac{2}{3} \times 3.92 = 2.35 \text{ tons on the square inch.}$$

SUPPLEMENTARY CALCULATIONS.—Factor q of Article 46.

$$\begin{array}{rcl} \text{Total sectional area,.....} & 1211 & \text{square inches.} \\ \times \text{Depth,.....} & 23.5 & \text{feet.} \\ \hline & 28458.5 & \text{product.} \\ \frac{7006}{28458.5} & = & 0.246, \text{ factor } q; \end{array}$$

being nearly the same as for a thin cylindrical tube.

Resistance to Horizontal Bending.—In the case of horizontal bending, the neutral axis is vertical and amidships; and the following calculation, based upon treating the ship as a rectangular tube, is nearly correct:—

	Area. Sq. inches.	Multipliers. Feet.	Products.
Sides,.....	440	17 (half-breadth)	7,480
Bottom, decks, and stringers, 771		17 ÷ 3	4,369
Moment of resistance to the modulus 1,.....			11,849 sum.
Greatest tension = Greatest thrust =	$\frac{27,470}{11,849} = 2.32$ tons on the sq. inch;		

being a very little less than the greatest thrust produced by bending in a vertical plane: which shows that the ship is a very little stronger horizontally than vertically.

The factor q has the following value for horizontal bending:—

Total sectional area,.....	1211 square inches.
× Breadth,.....	34 feet.
	41174 product.

$$\frac{11849}{41174} = 0.288, \text{ factor } q;$$

being intermediate between the values for a thin cylindrical tube and a thin square tube.

EXAMPLE II.—WOODEN SHIP.

STEP I.—Determination of neutral axis:—

	Area. Sq. inches.	Depths below B. Feet.	Products.
B,	1,955	0	0
E,	764	3.75	2,865
D,	1,994	7.50	14,855
F,	1,736	15.5	26,908
A,	5,216	23.5	122,576
11,665 total area,.....			167,204 sum.

Neutral axis below B,..... 14.3 feet = \overline{BC} .

Proportions in which the neutral axis divides the depth:—

\overline{AB}	:	\overline{BC}	:	\overline{CA}
:	:	23.5	:	14.3
:	:	23	:	14
:	:		:	9 very nearly.

STEP II.—Calculation of moment of inertia, and of moment of resistance to the modulus unity:—

	Area. Sq. inches.	Multipliers.	Products.
B,	1,955	(14.3) ³	399777.95
E,	764	(10.55) ³	85035.11
D,	764	(7.5) ³ ÷ 12	3581.25
D,	1,994	(6.8) ³	92202.56
F,	1,736	(1.2) ³	2499.84
F,	1,736	(16) ³ ÷ 12	37034.67
A,	5,216	(9.2) ³	488782.24

Divide by $\overline{BC} = 14.3$ )1058913.62 sum.

Moment of resistance to the modulus 1,..... 74050 nearly.

STEP III.—Calculation of the greatest stress, or actual modulus, with a known bending moment—

Divide by,.....	Greatest Bending Moment, as computed in Article 68.
	74050) 27470 foot-tons.

GREATEST TENSION,..... 0.371 ton on the sq. in.
(= 831 lbs. on the square inch);

And inasmuch as $\overline{CA} = \frac{9}{14} \overline{BC}$, we have—

GREATEST THRUST = $\frac{9}{14} \times .371 = .239$ ton on the square inch
(= 534 lbs. on the square inch).

SUPPLEMENTARY CALCULATIONS.—Factor q of Article 46.

Total sectional area,.....	11665 square inches
× Depth,.....	23.5 feet.
	274127.5 product.

$$\frac{74050}{274127.5} = 0.27, \text{ factor } q.$$

Resistance to Horizontal Bending.—Treating the ship, as before, like a rectangular tube, the following calculations are nearly correct:—

	Area. Sq. inches.	Multipliers. Feet.	Products.
Sides,.....	2,500	17 (half-breadth)	42,500
Bottom, decks, &c., 9,165		17 ÷ 3	51,935
Moment of resistance to the modulus 1,.....			94,435 sum.
Greatest tension = Greatest thrust =	$\frac{27470}{94435} = 0.291$ ton on the sq. in.		

(= 652 lbs. on the square inch);

being somewhat greater than the greatest thrust due to vertical bending, but still not above the safe working strength of timber.

The factor q has the following value for horizontal bending:—

Total sectional area,.....	11,665 square inches
× Breadth,.....	34
	396,610 product.

$$\frac{94435}{396610} = 0.238, \text{ factor } q;$$

being somewhat less than for a thin cylindrical tube.

77. *Proportionate Strength of different Vessels.*—The following three principles are convenient in their application to vessels made of the same materials distributed in the same way, and having similar cross-sections; but differing in absolute dimensions, and in the thickness of the skin, to which thickness the thickness of every piece that co-operates with the skin is supposed to be proportional. The skin, if made of iron, is also supposed to be stiffened by ribs or otherwise, so that its resistance to crushing is constant.

I. *The moment of resistance to bending is proportional to the breadth, the depth, and the mean thickness of the skin.*

Inasmuch as the greatest bending moment is proportional to the displacement and length, it follows that—

II. In vessels of the same material, similarly arranged, and of similar cross-sections, the mean thickness of the skin should be proportional directly to the load displacement and length, and inversely to the breadth and depth. This principle is the foundation of a rule deduced by Mr. J. R. Napier from the practical working of a great number of iron ships, viz:—

$$\left. \begin{array}{l} \text{Thickness of iron} \\ \text{skin in inches} \end{array} \right\} = \frac{\text{Displacement in tons} \times \text{Length in feet}}{800 \times \text{Breadth in feet} \times \text{Depth in feet}}$$

In vessels having similar longitudinal sections, as well as similar cross-sections, the displacement is proportional to the length, breadth, and depth; and therefore to make such vessels equally strong—

III. *The mean thickness of the skin should be proportional to the square of the length.*

But it must be observed, that in many cases of iron ships, especially those of small size, the preceding principles cease to be accurate, because of the modulus of the resistance of the skin to crushing by buckling being dependent on the proportion borne by the thickness of the skin to the distance (called *space*) between the ribs by means of which it is stiffened. In the example given in the preceding Article, and also in those to which Mr. J. R. Napier's rule, just cited, is applicable, the space is supposed not to be materially greater than about 30 times the thickness of the skin.

The best approximation to the resistance of an iron skin to buckling is that given by the Rule in Article 38 of this Division; but for proportions of space to thickness ranging from 35 to 80, the following rule, though not so precise, is simple and convenient, and its errors are on the safe side:—

$$\left. \begin{array}{l} \text{Ultimate strength against buckling, in} \\ \text{tons on the square inch} \end{array} \right\} = \frac{400 \times \text{thickness}}{\text{space}};$$

or with five as a factor of safety—

$$\text{Working modulus for thrust, in tons on } \left. \begin{array}{l} \text{the square inch} \end{array} \right\} = \frac{80 \times \text{thickness}}{\text{space}}.$$

Hence the following principles:—

IV. *The moments of resistance of iron ships in which the space between the ribs is 35 times the thickness of the skin and upwards, are proportional to the breadth, the depth, and the square of the thickness, and inversely proportional to the space; and consequently, in such ships—*

V. *The square of the mean thickness of the skin should be proportional directly to the displacement, length, and space, and inversely to the breadth and depth.*

In vessels having similar longitudinal sections as well as similar cross-sections, the displacement is proportional to the length, breadth, and depth; and therefore Principle V. takes the following form:—

VI. *The mean thickness of the skin should be proportional to the length and to the square root of the space.*

For a constant space, therefore, the thickness of the skin of an iron ship should be simply proportional to the vessel's length, and such is very nearly the ordinary practice; for the usual space between the frames of iron ships is 21 inches or thereabouts, and the mean thickness of the skin is very nearly $\frac{1}{35}$ part of an inch for each foot of the ship's length; or in other words, $\frac{1}{18\frac{1}{2}}$ of an inch for each 18 $\frac{1}{2}$ feet of the ship's length.

The thickness of the skin of an iron ship is slightly diminished towards the ends, because of the smaller bending moment. The thickness of the strakes is also made to diminish gradually in going upwards from the "garboard strakes," being those next the keel, until the "sheer strakes" are reached, which are again made somewhat thicker than those of the sides.

78. *Resistance to Longitudinal Racking* is given by the skin of the ship acting like the web of a girder (see Article 47 of this Division), and sometimes assisted by diagonal braces, which act like those of a skeleton beam (see Article 45 of this Division).

The *probable amount* of the racking force at the cross-sections where it is greatest, is given by the rules of Article 68 of the preceding Section; and the *probable greatest intensity* when borne by the skin, without the aid of diagonal braces, may be computed approximately by dividing that amount by the sectional area of those parts of the skin of the ship which act in the capacity of the web of a girder. In making that calculation, regard must be had to the fact, that the longitudinal racking force, like the bending moment, may act in an oblique longitudinal plane, and even almost parallel to the decks; and consequently the intensity of the racking stress is to be computed, regarding the duty of a web to be performed, first by the sides of the ship, and secondly, by her bottom.

The following example is taken from the iron ship of 2680 tons displacement already mentioned in the preceding Articles.

Greatest racking action, as calculated in Article 68, 429 tons.

Sectional area of skin of the two sides, regarded as a web, 298 square inches:—

$$\text{Greatest racking stress } \left. \begin{array}{l} \text{on the sides} \end{array} \right\} \frac{429}{298} = 1.44 \text{ ton on the square inch.}$$

Sectional area of skin of bottom, regarded as a web, 362 square inches—

$$\text{Greatest racking stress } \left. \begin{array}{l} \text{on the bottom} \end{array} \right\} \frac{429}{362} = 1.13 \text{ ton on the square inch.}$$

The resistance of the skin to racking stress is subject to a deduction of about $\frac{1}{10}$ for the rivet-holes in the seams. If we take, as in previous examples, 4 tons on the square inch as the working modulus for tension and for racking stress on solid metal, $4 \times 0.7 = 2.8$ tons will be the working modulus for racking stress, being more than double of the greatest actual racking stress; so that, in the example chosen, there is a very great surplus of strength against longitudinal racking in the skin alone, without the aid of diagonal braces; and such is the case in most iron-skinned ships.

In wooden-skinned ships longitudinally planked, and without diagonal braces, the racking force tends to make the strakes of plank slide upon each other at their seams; and the mutual friction and adhesion of the edges of the planks are not sufficient to resist that force permanently, as is shown by the arching or "hogging" of such ships. It would appear, however, from experience, that the friction and adhesion at the seams are capable of resisting a large proportion of the racking force; for the quantity of diagonal bracing which is found to answer in practice is by no means sufficient of itself to bear the whole racking force with safety.

For example, the wooden ship of 2680 tons, already referred to, if diagonally braced in conformity with the rules of "Lloyd's Register of Shipping," has each cross-section of greatest racking force traversed by six flat iron diagonal bars (three at each side between the frame and skin), sloping at about 45°, and each measuring 5 inches by $\frac{3}{4}$ inch. We have then—

Total area of braces at each cross-section, $5 \times \frac{3}{4} \times 6 = 22\frac{1}{2}$ square inches.

These bars are in general of stronger iron than the skin of the ship; and taking 25 tons on the square inch as their ultimate tenacity, and 5 as the factor of safety, their working modulus, for tension, is found to be 5 tons on the square inch. Then—

	Tons.
Total safe tension on the braces = $22\frac{1}{2} \times 5 = \dots\dots\dots$	112.5
$\times \sin 45^\circ = \dots\dots\dots$	79.7
Vertical component of that tension, $\dots\dots\dots$	79.5
Greatest racking force, as before, $\dots\dots\dots$	429.0
Of which there remains, to be resisted by the wooden skin, $\dots\dots\dots$	349.5

The diagonal bracing of the bottom and decks, similar to that of the sides, helps in the same manner the planking to resist that part of the racking force which acts in a horizontal plane.

Wooden-skinned ships have been built with three layers of planking; the two inner layers running diagonally in opposite directions, and the outer layer longitudinally. In such cases, the sectional area of *both diagonal layers* may be considered as available to resist racking force, and the sectional area of the longitudinal layer, added to *one-half* of that of the diagonal layers, as available to resist bending moment.

In providing a ship with diagonal tie-braces, regard must be had to the direction of the most severe racking force. Thus, most wooden ships tend to hog, or arch upwards in the middle, and droop at the ends; hence the iron diagonal braces slope *downwards* from the middle towards the ends. Should a ship tend to sag in the middle, the diagonal braces, so far as the sagging action extends, should slope *upwards* from the middle. The diagonal braces should be strongest and most numerous at the cross-sections of greatest racking force.

79. The *Transverse Bending Action*, described in Article 69 of this Division, as being most severe when the vessel takes the ground and is supported wholly on the keel, is resisted partly by means of the beams tying the sides together, and partly by the transverse strength of the *floors*, which are the lowest parts of the ribs or frames, crossing the keel at right angles.

In iron-framed ships, each of the floors usually consists of a plate-iron web, called the "floor-plate," with angle-iron flanges rivetted upon its upper and lower edges, so as to form an I-shaped beam; and they are made so deep and so stiff, compared with the other parts of the frames and with the beams, that they may be considered as practically bearing the *whole* of the transverse bending moment at the points where they cross the keel.

In wooden-framed ships, the floors are square pieces of timber, of dimensions somewhat, though not very much greater, than those of the other pieces of the frames to which they belong, and of the beams; and the beams and pieces of the frames are so connected together, that about *one-half* of the transverse bending moment may for practical purposes be regarded as borne by the floors where they cross the keel, the other half being borne by means of the tension of the deck-beams.

EXAMPLE I.—Iron ship.

Displacement,	2680 tons
× mean breadth, about.....	26 feet.
Divide by.....	8) 69680 product.
Total bending moment,.....	8710 foot-tons.
Total bending moment 8710 foot-tons = { 42·5 foot-tons per foot, or	
Length of ship 205 feet = { 42·5 inch-tons per inch,	
× "Room and space," say.....	24 inches.
Bending moment on each floor,.....	1020 inch-tons.

Suppose the floor to consist of a web 24 inches deep by $\frac{1}{8}$ of an inch thick, with double angle-iron flanges along its edges, each of 6 square inches sectional area; then we have the following calculation for the moment of resistance to the modulus 1:—

	Areas.	Mean Leverages.	
Flanges,.....	6 × 2	× 24 ÷ 2	= 144
Web,.....	24 × $\frac{1}{8}$	× 24 ÷ 6	= 66
			210 sum.

Then the greatest stress is found to be $\frac{1020}{210} = 4·9$ tons on

the square inch; and if we take the modulus of rupture of the iron at 25 tons on the square inch, the factor of safety is found to be about 5, which is quite sufficient.

It is to be observed, that the floors are more likely to give way by tension at the upper edge than by compression at the lower edge, because of the stiffness and strength given to the lower flange by its connection with the ship's skin.

Iron ships are sometimes built with what is called a "deep-plate keel," in which the keel has a longitudinal web cutting the floor-plates in two. It is evident that in such cases the two halves of each floor ought to be tied together across the keel with bars or bolts of strength sufficient to make up for the loss of the strength of their webs or floor-plates.

EXAMPLE II.—Wooden ship.

Bending moment per inch of length as in Example I.,	42·5 inch-tons
× Room and space, say	38½ inches.
Bending moment per frame,.....	1423·75 inch-tons.
One-half of the above, borne by each floor where it crosses the keel,.....	711·875 inch-tons.

For a floor "sided 15 inches, moulded 15 inches," that is, 15 inches broad and 15 inches deep, the moment of resistance to the modulus 1 is—

$$15 \times 15^2 \div 6 = 562·5;$$

consequently the greatest stress is—

$$\frac{711·875}{562·5} = \begin{cases} 1·266 \text{ tons on the square inch, or} \\ 2936 \text{ lbs. on the square inch.} \end{cases}$$

This shows the factor of safety to be between 4 and 5; which is enough for a dead load on timber, but not for a live load. The ship therefore may be expected to rest with safety on her keel, provided she settles down upon it gradually and without shock or vibration, but not otherwise.

It is here to be remarked, that according to one mode of building wooden ships, the keel is crossed at each station by a *pair* of floors of equal size, bolted together side by side; but the strength of only one floor of each such pair is reckoned as available, because of the butt joints at which they terminate to one side and to the other side of the keel alternately. According to another mode of building, a pair of "half-floors" meet at a butt joint above the keel, and are fished together by means of a "cross-piece" which lies alongside of both; and then it is the transverse strength of the cross-piece that resists the transverse bending action.

The effect of *pillars* or *stanchions*, transmitting vertical pressure from the deck-beams to the keelson, is to enable the transverse strength of the deck-beams partially to assist that of the floors in bearing the transverse bending action. To find how much is to be added to the resistance of a floor on account of that of a beam with which it is thus connected, compute the moment of resistance of the beam to the modulus 1 in the usual way, and multiply it by the ratio which the depth of the beam bears to the depth of the floor; then add the product to the moment of resistance of the floor to the modulus 1.

In iron ships, the depth of the beams is usually so small compared with that of the floors, that but little assistance can be given to the floors in the manner just mentioned. In wooden ships it is otherwise, as the following example illustrates:—

EXAMPLE III.—A floor 15 inches square has a beam 14 inches square propped by a stanchion directly above it. In what proportion is the resistance to bending increased?

Moment of resistance of the floor to the modulus 1, as calculated in Example II.,	562·5
Addition on account of the beam, $\frac{14 \times 14^3}{6} \times \frac{14}{15} =$	426·8
	989·3 sum.

Proportion in which the strength is increased—

$$\frac{989·3}{562·5} = 1·76 \text{ nearly;}$$

and if every frame were strengthened in the same way, the factor of safety would be between 7 and 9 instead of between 4 and 5.

As the strength of floors of similar cross-section increases proportionally to the breadth (or "siding") and square of the depth (or "moulding"), it is obvious that in whatsoever proportion the moulding of the floors of a ship is increased (preserving their form of section), the ratio in which the room and space is greater than the siding of the floors may be increased in the *square* of that proportion, and the strength against transverse bending will still be the same, while material will be saved; care being taken not to increase the room and space more than is consistent with proper

stiffening of the skin. For example, in the wooden ship already cited, suppose the siding of the floors to be still 15 inches, but the moulding to be increased to 18 inches, that is, in the proportion of 6 to 5; then the room and space may be increased in the ratio of 36 to 25; that is, instead of $33\frac{1}{2}$ inches, it may be made—

$$\frac{33\frac{1}{2} \times 36}{25} = 48\frac{1}{2} \text{ inches, nearly.}$$

The quantity of timber in the floors will, at the same time, be diminished to $\frac{5}{6}$.

When a ship is divided into compartments by *plate-iron bulkheads*, firmly connected with the framework and skin, those contribute to resist the transverse bending action; and should the skin between them be properly stiffened by means of stringers or longitudinal ribs, to be afterwards more fully described, they may bear the whole of that action.

Each bulkhead should be sufficiently stiffened by means of angle-iron or T-iron ribs, running along and across the seams of the plating of which it is built; and the modulus of resistance to buckling may be computed from the ratio which the thickness of the plating bears to the space between those ribs, by the rule of Article 38 of this Division, or more simply, though less exactly, by that applied to the skin in Article 78. Suppose, for example, as is not uncommon, that the space between the ribs is 50 times the thickness of the plating; then the *working modulus of resistance to buckling* is, by the rule of Article 38, with the factor of safety 5—

$$\left(1 + \frac{2500}{3000}\right) \times 5 = 3960 \text{ lbs.} = 1.77 \text{ tons on the square inch;}$$

by the rule of Article 78, $\frac{80}{50} = 1.6$ tons on the square inch.

The stiffening ribs may be taken as increasing the *effective sectional area* of the bulkhead, regarded as a beam, nearly in the same proportion in which they add to its weight; that is, about *one-fifth*.

Owing to the great depth of a bulkhead, the *racking stress* connected with the transverse bending action may often be very considerable; and it should be computed, by dividing *half the displacement* of the vessel by *two-thirds of the combined effective sectional areas* of all the bulkheads.

The firm connection of the bulkhead with the bottom of the ship provides it with a lower stringer or flange of such stiffness, that its giving way to the bending moment must take place by tension, if at all; and the working modulus of resistance to this action may be taken at 4 tons per square inch.

The stem and stern of the vessel may be regarded as together doing the duty of one bulkhead in resisting the racking and bending actions now in question.

EXAMPLE IV.—*Question 1.*—What amount of racking force, and what transverse bending moment, can safely be borne by a bulkhead 20 feet deep and $\frac{5}{8}$ of an inch thick, with the working modulus of stress 1.6 ton on the square inch against buckling, and 4 tons on the square inch against tearing?

			Squares.
Depth 240 in. \times thickness $\frac{5}{8}$ in. =			150
Add for stiffening ribs, one-fifth, =			30
			180
$\frac{5}{8} \times 180 \times 1.6 = 192$ tons, safe racking load.			
Eff. Area.	Depth.	Modulus.	
180 \times	240 \times	4 =	{ 28,800 inch-tons =
	6		{ 2,400 foot-tons, safe bending moment.

Question 2.—What is the least number of such bulkheads that would safely bear a racking load equal to half the displacement of the vessel referred to in the preceding examples, and the bending moment already calculated?

$$\frac{\text{Half-displacement } 1340}{192} = 7;$$

$$\frac{\text{Moment as computed } 8710 \text{ foot-tons}}{2400} = 3.63;$$

then deducting 1 from the greater of those numbers for the stem and stern, it appears that *six* bulkheads is the number required. These would divide the ship into seven compartments, each about 29 feet long.

81. *Resistance to Transverse Compression.*—The mode of action of transverse compression has been explained and illustrated, and the computation of its amount exemplified, in Article 70 of the present Chapter, and Figs. 4, 5, and 8. It is opposed partly by the resistance of the ribs of the vessel to disfigurement, and partly by the direct resistance of floors, beams, and bulkheads to compression.

Each rib, together with the beam or beams with which it is connected by means of knees, forms a sort of stiff ring or hoop, which the transverse compression tends to make more curved at its highest and lowest parts, and less curved at its lateral parts. The greatest bending moment which thus acts alternately outwards and inwards at different parts of the ring, may be determined, for different forms of frame in different positions, by a method of investigation which is too intricate to be given here in detail, but which leads to this result—that for the ordinary forms of vessels, the greatest bending moment produced by transverse compression does not greatly differ from *the amount of the transverse pressure, acting with a leverage equal to one-fifteenth part of the greatest depth of immersion of the surface to which that pressure is applied.*

EXAMPLE I.—Depth of immersion of the surface considered, as in Article 70, $17\frac{1}{2}$ feet;

Amount of transverse pressure, as computed in Article 70, per foot of length, 4.375 tons;

$$\text{Bending moment, } \frac{4.375 \times 17\frac{1}{2}}{15} = \begin{cases} 5.1 \text{ foot-tons per foot of room and space, or} \\ 5.1 \text{ inch-tons per inch of room and space.} \end{cases}$$

This moment acts with most severity upon the frames at and near the bilges. In an ordinary iron ship, each frame, at that part of the vessel, consists of a pair of angle-irons rivetted together in reversed positions so as to form a sort of Z-shaped section, which may be regarded as practically equivalent to an I-shaped section; and as the outermost of those angle-irons is rivetted to the skin of the ship, the part of the skin extending to a breadth of half the room and space to either side of the frame may be considered as forming part of one of the flanges of the rib. The dimensions of the frames of an iron ship are seldom sufficient to resist the transverse compression unaided by bulkheads; unless they are those frames with very deep webs, placed at large intervals apart, to which Mr. Scott Russell gives the name of “partial bulkheads.”

In an ordinary wooden ship, the part of each frame on which the moment of transverse pressure acts with most severity is usually what is called the “*second futtock*,” and is of somewhat less siding and moulding than the floor-timber.

EXAMPLE II.—Suppose in an iron ship, subjected to the action computed in Example I., that the room and space is 24 inches;

and that each frame at the bilge is equivalent to an I-shaped beam of the following dimensions:—

Depth,.....	5 inches.
Area of inner flange,.....	2 sq. inches.
Area of web,.....	4 sq. inches.
Area of outer flange (including $24 \times \frac{3}{8} = 19.5$ square inches of skin),.....	22 sq. inches.
Total area,.....	28 sq. inches.
Bending moment per frame,.....	$5.1 \times 24 = 122.4$ inch tons.
Moment of resistance to the modulus 1 (calculated by Case X. of the Table under Rule IV. of Article 46 of this Division), area, 28 sq. inches \times depth, 5 inches \times factor g , 0.124.....	$= 17.36$
Then we find greatest stress, $\frac{122.4}{17.36} = 7$ tons on the square inch;	

being a safe stress under a dead load, but not under a live load; therefore the frames will eventually bulge either inward or outward (generally inward) if left to bear the transverse pressure unassisted.

EXAMPLE III.—In a wooden ship, subjected to the action computed in Example I., let the room and space be $33\frac{1}{2}$ inches, and the second futtocks sided and moulded $12\frac{3}{4}$ inches. Then—

$$5.1 \times 33\frac{1}{2} = 270.85 \text{ inch-tons, moment per frame;}$$

$$\frac{(12.75)^3}{6} = 345.5, \text{ moment of resistance to the modulus 1;}$$

$$\text{Greatest stress} = \frac{270.85}{345.5} = \begin{cases} 0.784 \text{ ton on the square inch, or} \\ 1756 \text{ lbs. on the square inch;} \end{cases}$$

so that the factor of safety for oak or teak is between 7 and 8.

When the transverse pressure is to be resisted by *plate-iron bulkheads*, it is necessary that there should be longitudinal frames or stringers of strength sufficient to transmit to the bulkheads the pressure against those parts of the skin which lie between them. The strength of those stringers will be considered in the next Chapter.

In this case the transverse pressure is resisted partly by the bulkhead, partly by the beams, and partly by the skin and transverse framing of the bottom of the vessel; and when the material is distributed in the usual way, the intensity of the stress may without practical error be treated as uniform throughout those pieces of material.

The resistance of the bulkheads to buckling may be estimated as in Article 79, Example IV.

These principles are illustrated in the following calculation:—

EXAMPLE IV.—Suppose an iron ship, without floor-plates, and having the following dimensions and arrangement of materials—

Depth of immersion,.....	20 feet.
Interval between bulkheads,.....	29 "
Thickness of bulkheads,.....	$\frac{5}{8}$ inch.
Ribs, &c., of bulkheads equivalent to an additional thickness of,.....	$\frac{1}{8}$ "
Room and space,.....	24 inches.
Two decks, with a deck-beam in each deck to each alternate frame; sectional area of beam,.....	9 sq. inches.
Sectional area of each transverse frame of the bottom, .	$7\frac{1}{2}$ "
Mean thickness of plating of bottom,.....	$\frac{3}{8}$ "

Then the calculation of stress due to transverse pressure is as follows:—

Transverse pressure on each length of 29 feet—

$$\frac{29 \times 20^2}{35 \times 2} = 166 \text{ tons.}$$

Resisting area in each length of 29 feet—

		No.	Square Inches.
Beams,.....	$9 \times$	$14\frac{1}{2}$	$= 130.50$
Frames,.....	$7\frac{1}{2} \times$	$14\frac{1}{2}$	$= 108.75$
Bottom Plating,.....	$\frac{1}{8} \times 29 \times 12$		$= 282.75$
Bulkhead,.....	$240 \times$	$\frac{3}{4}$	$= 180.00$
Total,.....			702.00

$\frac{166}{702} = 0.236$ ton per square inch, intensity of transverse thrust; being a trifling stress in comparison with that due to the transverse bending, as computed in the preceding Article.

Reference has already been made to *partial bulkheads*, or deep-webbed frames, placed at large intervals apart. Such frames, like complete bulkheads, require a system of longitudinal frames to transmit to them the pressure against the intermediate part of the skin.*

As in the case of ordinary frames, each partial bulkhead, together with the division of the skin to which it is fastened, may be regarded as forming an I-shaped girder, with a very broad flat flange at its outer edge.

EXAMPLE V.—Iron Ship:—

Greatest immersion,.....	13 feet.
Interval between partial bulkheads,.....	13 "
Thickness of skin at bilge,.....	$\frac{5}{8}$ inch.
Depth of web of partial bulkhead,.....	13 inches.
Thickness " " " ".....	$\frac{3}{4}$ inch.
Sectional area of angle-irons along edges of web, each... 2.75 sq. inches.	

From these data the following results are computed—

$$\text{Transverse pressure,.....} \frac{13^2}{2 \times 35} = 2.414 \text{ tons per foot of length.}$$

$$\text{Bending moment at bilge,.....} \left\{ \frac{2.414 \times 13 \text{ ft. (immersion)}}{15} = 2.092 \right\} \text{ foot-tons per foot of length}$$

$$\times \text{Interval between partial bulkheads,.....} 13$$

$$\text{Bending moment on each partial bulkhead,.....} \left\{ \begin{array}{l} 27.196 \text{ foot-tons.} \\ \times 12 \\ \hline 326.352 \text{ inch-tons.} \end{array} \right.$$

Moment of resistance to the modulus 1, calculated by Case X. of the Table under Rule IV. of Article 46 of this Division.

$$\text{Area, } 112.75 \text{ sq. in.} \times \text{depth, } 13 \text{ in.} \times \text{factor } g, 0.05217 = 76.47.$$

$$\text{Greatest stress, } \frac{326.352}{76.47} = 4.27 \text{ tons on the sq. inch;}$$

so that the factor of safety is between 5 and 6, according to the quality of the iron in the partial bulkheads, and is amply sufficient.†

* For an example of this method of building, see a paper by Mr. Scott Russell "On the Longitudinal System in the Structure of Iron Ships," published in the Transactions of the Institution of Naval Architects for 1862.

† The data of this example are taken from the description of the *Annette*, in Mr. Scott Russell's paper already mentioned, except the depth of immersion, which is estimated at about 7-10ths of the depth in hold.

CHAPTER III.

OF THE STRENGTH OF PARTICULAR PARTS AND EQUIPMENTS OF SHIPS.

82. *Local Stress upon the Skin.*—In what direction soever the frames run by which the skin is stiffened, the part of the skin between each pair of frames is in the condition of a *continuous beam* (see Article 53 of this Division, Fig. 14), of which the two frames correspond to the props, and the “room and space” to the span, while the load, tending to bulge the skin inwards between the frames, and make it concave inwards at the frames, consists of the pressure of the water.

Consider a strip of skin one inch broad. The load on that strip is as follows, in *tons* :—

$$\frac{\text{Depth of immersion in feet} \times \text{room and space in inches}}{5040 (= 12 \times 12 \times 35)}$$

According to the principles explained in Article 53, just referred to, the greatest bending moments produced by that load on the strip of skin are, in *inch-tons*—

$$\text{At the frames, } \frac{\text{load} \times \text{room and space}}{12};$$

$$\text{Midway between the frames, } \frac{\text{load} \times \text{room and space}}{24}$$

The *stress* produced amongst the particles of the skin by the action now described, has the following value, in *tons on the square inch*—

$$\frac{6 \times \text{bending moment}}{\text{square of thickness of skin}}$$

Although the stress thus produced is greatest at the points where the skin crosses the frames, its effects are of most importance at the parts midway between the frames; because it is at the latter points that the skin tends to give way, by buckling under the thrust produced by the bending of the ship as a whole. Combining, then, the expressions already given, we find, for the stress in tons on the square inch, amongst the particles of the skin midway between the frames, due to the local pressure of the water, the following value—

$$\frac{6 \times \text{Depth of immersion in feet} \times (\text{room and space in inches})^2}{5040 \times 24 \times (\text{thickness of skin})^2} \\ = \frac{\text{Depth of immersion in feet} \times (\text{room and space in inches})^2}{20160 \times (\text{thickness of skin})^2}.$$

EXAMPLE I.—In an iron ship, let the depth of immersion of the bottom be 20 feet, the room and space 24 inches, and the thickness of the bottom-plating 13-16ths of an inch; then the stress in question is—

$$\frac{20 \times 24^2 \times 16^2}{20160 \times 13^2} = 0.86 \text{ ton on the square inch.}$$

EXAMPLE II.—In a wooden ship, let the depth of immersion of the bottom be 20 feet, the room and space 33½ inches, and the thickness of the bottom planking 4½ inches; then the stress in question is—

$$\frac{20 \times (33\frac{1}{2})^2}{20160 \times (4\frac{1}{2})^2} = \begin{cases} 0.055 \text{ ton on the square inch, or} \\ 123 \text{ lbs. on the square inch.} \end{cases}$$

If we take the stress on the particles of the bottom due to

longitudinal bending, as already computed in Article 77 of the preceding Chapter, and combine it with that just calculated, the results are as follows :—

	Iron Ship.		Wooden Ship.	
	Tons on the square inch.		Tons on the square inch.	Lbs. on the square inch.
Stress due to longitudinal bending,....	2.35	0.239	= 535
“ local pressure,.....	0.86	0.055	= 123
Sum,.....	3.21	0.294	= 658

83. *Local Stress upon Longitudinal Ribs—Breasthooks and Transoms.*—It has been already stated in Article 81, that when a ship's skin is stiffened by means of deep frames, or of complete or partial bulkheads, at long intervals apart, longitudinal ribs or stringers are required to transmit the pressure upon the skin to the transverse ribs or bulkheads. In an iron ship, the best arrangement for such stringers appears to be, one along the centre of each strake of the plating,^c consisting of a web with an angle-iron flange along each edge, the outer flange rivetted to the strake of plating.

Each stringer, together with the strake to which it is fastened, composes virtually an I-shaped continuous beam, supported at the complete or partial bulkheads, and loaded with the pressure of the water. The greatest bending moments are computed as follows, in *inch-tons*—the depth of immersion being in *feet*, and the breadth and interval in *inches* :—

Midway between the supports, producing convexity inwards—

$$\frac{\text{Immersion} \times \text{breadth of strake} \times \text{interval}^2}{5040 \times 24 (= 120960)};$$

At the supports, producing concavity inwards—

$$\frac{\text{Immersion} \times \text{breadth of strake} \times \text{interval}^2}{5040 \times 12 (= 60480)}.$$

The neutral axis is so near the skin, that the stress produced on the skin is trifling. The most severe stress is exerted at the inner edge and flange of the web. Midway between the bulkheads it is *tensile*, and is therefore *opposed* to the thrust exerted through the same particles in consequence of the longitudinal bending of the whole ship. At the bulkheads it is *compressive*, and is *added* to the thrust due to the longitudinal bending of the whole ship. The latter, therefore, is the stress to be calculated; and it is found by dividing the bending moment as given in the second of the preceding formulæ, by the moment of resistance to the modulus 1 of the beam formed by the strake and longitudinal rib, as computed by Case XI. of the Table under Rule IV. of Article 46 of this Division.

EXAMPLE.

Depth of immersion,.....	13 feet.
Breadth between longitudinal ribs,....	36 inches.
Interval, or span,.....	13 feet = 156 inches.
Thickness of skin,.....	¾ inch.
Web of longitudinal rib,.....	18 inches × ½ inch.
Angle irons,.....	3½ inches × 3½ inches × ½ inch.

^c See Mr. Scott Russell's description of the *Annette*, in the paper already cited (Transactions of the Institution of Naval Architects, 1862).

From these data the following results are computed:—

$$\text{Bending moment at points of support, } \frac{13 \times 36 \times 156^2}{60480} = 188 \text{ inch-tons.}$$

$$\begin{array}{l} \text{Area.} \quad \text{Depth.} \quad \text{Factor } q. \\ \text{Moment of resistance to the Modulus } 1, \dots 26.75 \times 18 \times 0.248 = 119. \\ \frac{188}{119} = 1.58 \text{ tons on the square inch, intensity of thrust along the inner edge of} \\ \text{the longitudinal rib, where it crosses the bulkheads.} \end{array}$$

The intensity of the thrust along the bottom-plating, due to longitudinal bending of the ship as a whole, computed to a rough approximation, is about 3 tons on the square inch. The inner edge of each longitudinal rib of the bottom is 18 inches nearer the neutral axis of the ship than the plating of the bottom—that is, at about $\frac{2}{3}$ of the distance of the bottom from that axis; therefore the thrust along it is about $\frac{2}{3}$ of the thrust along the bottom-plating, or $2\frac{2}{3}$ tons on the square inch. Adding those thrusts together, we find about 4 tons on the square inch for the total intensity of the thrust. That stress takes place at a point where the rib is completely stiffened laterally, and where, consequently, its ultimate resistance to compression may be estimated at about $16\frac{1}{2}$ tons to the square inch, nearly; therefore the factor of safety is somewhat more than 4.

The longitudinal ribs of the sides may be made smaller than those of the bottom, as having less pressure to bear (for example, in the *Annette* they are 13 inches deep instead of 18); but it is to be observed, that those of the bottom may often be almost entirely relieved from local stress, through the weight of the cargo resting upon them, and directly opposing the pressure of the water; while those of the sides are always exposed to pressure from without.

When an iron ship has an *inner skin*, it may be regarded as forming inner flanges to the longitudinal as well as the transverse ribs.

Breasthooks at the bow, and *Transoms* at the stern (shaped and placed as described in the Second Division, Article 29) are either nearly or exactly in the position of longitudinal frames, and their strength depends on the same principles. By far the most efficient kind of framing for the bow, even of a ship which is transversely framed elsewhere, consists of a series of breasthooks, in planes perpendicular, or nearly so, to the *stem*, and running back to the forward bulkhead, where their after ends are supported. For an example of this, see the longitudinal section of the *Persia*, Plate $\frac{4}{5}$.

84. *Local Strength of Keelsons, Keels, and Floors*.—When the word *keelson* is used in the singular without qualification, it means a longitudinal piece, parallel to and directly above the keel. Other longitudinal pieces in the framework of the ship's bottom are called *sister keelsons*, and if they are in the bilge, *bilge keelsons*.

In wooden-framed ships, the keelsons all run along inside the transverse frames. In iron-framed ships, they run sometimes inside the transverse frames, and sometimes in the spaces between them. In the latter case they are called *intercostal keelsons*, and are firmly fastened to the frames and to the skin by angle-irons and rivets.

The position and use of intercostal keelsons exactly resemble those of the longitudinal frames mentioned in Article 83, from which they are distinguished only by consisting of shorter pieces.

All keelsons whatsoever act along with the skin, and with the other longitudinal pieces of the framework, in resisting the longitudinal bending of the ship as a whole.

The local stress which keelsons are specially designed to bear (and in bearing which the main or midship keelson is more or less

assisted by the keel), is that arising from their use to *distribute* over the framing of the ship's bottom any too much concentrated load, such as the thrust of the masts, or the weight and heaving reaction of the engines and boilers. Keelsons used for this purpose are sometimes called *Sleepers* or *Bearers*.

To find approximately the bending moment upon a keelson under such circumstances, *multiply the load by one-eighth part of the difference between the length on which the load is concentrated, and the length over which it is to be distributed.*

EXAMPLE.—Distribution of the weight and heaving reaction of a pair of engines. Suppose that the weight of a pair of engines, 400 tons, is concentrated upon a length of 30 feet of the ship's floor, and is to be distributed over 50 feet by means of four parallel wrought iron hollow rectangular or "box" keelsons; depth of each keelson, 36 inches; breadth, 18 inches; thickness of the top and bottom to be double that of the sides, so that the factor q is the same as for a thin hollow square, viz., $\frac{1}{3}$ (see Case V. of the Table under Rule IV. of Article 46); working modulus of strength, 4 tons on the square inch:—

Weight,.....	400 tons.
Add for reaction due to the heaving of the waves, $\frac{1}{4}$,.....	100 tons.
Total load to be distributed,.....	500 tons.
Multiply by difference of lengths, 50—30 = 20	
Divide by,.....	8) 10,000
	1250 foot-tons, moment;
	× 12
Divide by number of keelsons,.....	4) 12,000 inch-tons.
Divide by working modulus,.....	4) 3000 {inch-tons, bending mo-
Divide by $\frac{1}{3}$ of depth,.....	12) 750 {ment on each keelson.
	Required moment of re-
	{istance to modulus 1.
Required sectional area of iron in each keelson, 62 $\frac{1}{2}$ square inches.	

Supposing the whole available section to consist of plates, this area would be given by a thickness of 0.434 inch for the sides, and 0.868 inch for the top and bottom.

85. *Local Strength of Decks*.—A deck of the usual construction consists of a *flat*, or platform, of planking, and sometimes of iron plate, supported by *beams*. Timber beams are rectangular, and in general square; iron beams are of a T-shaped or I-shaped section, and so proportioned that the factor q is about one-third of the depth. The beams are rigidly connected, by means of *knees*, to the frames of the ship's sides. The knees in iron ships are of iron, and their place is sometimes supplied by *bracket ends* on the beams. In wooden ships, they are of iron or wood. The ends of the beams in wooden ships rest on a strong longitudinal piece called a *shelf*, and are also secured in their places by a longitudinal piece above them called a *water-way*. The shelf and waterway are both fastened to the frames of the side.

In iron ships, each deck has usually a beam to each alternate frame, so that the *spacing* of the beams is from $3\frac{1}{2}$ to 4 feet; in wooden ships, the ordinary spacing is about 4 feet or $4\frac{1}{2}$ feet. In ships having a deep hold, there is sometimes a tier of beams called *hold-beams*, not carrying a deck, but contributing to the resistance to transverse pressure.

The manner in which a deck contributes to the strength of the ship as a whole has been described in the preceding Chapter. The local stress upon a deck is that produced by the weight and heaving reaction of the direct load upon it.

According to the rules commonly followed in practice as to the

dimensions of deck beams, the *moulding*, or depth, of *iron beams* is $\frac{1}{8}$ part of their length; the thickness of the web, $\frac{1}{16}$ of the depth; and the flanges are together of an area about equal to that of the web.

The moulding of wooden beams ranges from about $\frac{1}{8}$ to $\frac{1}{6}$ of their length, and they are usually square. The "round-up" of the deck is often given by diminishing the depth at the ends to about $\frac{5}{8}$ of the depth in the middle—a practice injurious at once to general and to local strength.

The rigid connection of the beams with the sides puts them in the condition of beams partially, but not absolutely, fixed at the ends; and the greatest bending moment may be estimated as being nearly equal to the *total load* (supposed uniformly distributed) multiplied by $\frac{1}{16}$ part of the length.

EXAMPLE I.—Given, for an iron deck-beam—

Length,.....	36 feet,
Depth,.....	9 inches,
Sectional area,.....	10·125 square inches,
Working modulus of strength,.....	5 tons on the square inch.

Required the greatest safe uniformly distributed working load :—

$$\text{Working moment of resistance, } \frac{5 \times 9 \times 10 \cdot 125}{8} = \begin{cases} 151 \cdot 875 \text{ inch-tons, or} \\ 12 \cdot 656 \text{ foot-tons.} \end{cases}$$

Divide by $\frac{1}{16}$ of length—

$$\frac{12 \cdot 656}{2\frac{1}{2}} = 5 \cdot 625 \text{ tons, greatest safe working load;}$$

or very nearly $\frac{1}{2}$ of a ton per foot.

If the beams are spaced 4 feet, this is equivalent to 87 lbs. on each square foot of deck; being the weight of about $16\frac{1}{2}$ inches depth of sea-water.

EXAMPLE II.—Given, for a wooden beam—

Length,.....	32 feet,
Siding and moulding,.....	12 inches,
Working modulus,.....	$\frac{1}{2}$ ton on the square inch.

Required the greatest safe uniformly distributed working load :—

$$\text{Working moment of resistance, } \frac{1}{2} \times \frac{12^3}{6} = 144 \text{ inch-tons} = 12 \text{ foot-tons.}$$

Divide by $\frac{1}{16}$ of length—

$$\frac{12}{2} = 6 \text{ tons, greatest safe working load;}$$

or 0·1875 ton per foot of length.

If the beams are spaced 4 feet, this is equivalent to 105 lbs. on each square foot of deck; being nearly the weight of 20 inches depth of sea-water.

By propping beams amidships with *stanchions*, which transmit part of the load to the middle-line keelson, they are made about one-half stronger.

The flat of the decks, when of timber, is usually from $2\frac{1}{2}$ to 4 inches thick, according to the size of the vessel. The chief stress upon it is that arising from the longitudinal bending of the whole ship, discussed in the preceding Chapter. When it consists of iron plates, they are made about half the mean thickness of the ship's skin.

86. *Strength of the Rudder and its Supports*.—The most severe stress on a common rudder is that which arises from the twisting action on the head, when it is put hard over. In Division I., Article 188, a rule, founded on experiments by Mr. J. R. Napier, has been given for estimating the pressure on the rudder, when put over to an angle of 40° , and the twisting moment (see Transactions of the Institution of Naval Architects, 1864). That rule,

when modified so as to give the results in *tons*, and in *inch-tons*, takes the following form :—

I. For the pressure in tons, when put over to 40° , multiply the immersed area of the rudder in square feet by the square of the velocity, in knots, with which the water passes it, and divide by 2400. At angles different from 40° , the pressure may be taken as nearly proportional to the square of the sine of the angle.

II. For the twisting moment in inch-tons, multiply the pressure in tons by the distance in inches of the centre of the immersed area of the rudder from its axis of motion.

III. The moment of resistance of the rudder-head according to the principles of Article 58 of this Division, is equal to

Working modulus of strength $\times 0 \cdot 196 \times \text{cube of diameter in inches}$;

And consequently—

IV. To find the diameter of the rudder-head in inches, multiply the twisting moment when the helm is hard over by $5 \cdot 1 = \left(\frac{1}{0 \cdot 196}\right)$; divide the product by the working modulus, and extract the cube root of the quotient.

The values of the working modulus in ordinary practice appear to be as follows :—

For iron,.....	$3\frac{1}{2}$ tons on the square inch.
For timber,.....	$\frac{1}{2}$ ton on the square inch.

Giving factors of safety of about 6 for iron (large forgings), and 8 for timber.

A timber rudder-head is almost exactly *three times* the diameter of an iron rudder-head of the same strength.

EXAMPLE.—Given: immersed area of rudder, $20 \times 4\frac{1}{2} = 90$ square feet; velocity of water passing it, 12 knots; distance of centre of immersed area from axis of motion, 27 inches; required, the diameter of the iron rudder-heads.

Pressure,.....	$\frac{90 \times 12^2}{2400} = 5 \cdot 4 \text{ tons;}$
	$\times 27 \text{ inches leverage}$
Twisting moment,.....	$145 \cdot 8 \text{ inch-tons}$
	$\times 5 \cdot 1$
Divide by working modulus,.....	$3 \cdot 5 \quad 743 \cdot 58 \text{ product}$
Cube of diameter,.....	$212 \cdot 31$
	$\sqrt[3]{212 \cdot 31} = 5 \cdot 966 \text{ inches, diameter required.}$

The *bending moment on the tiller*, or on a *yoke* of which one arm only is strained at a time, is equal to the twisting moment on the rudder-head; but if the force for putting the helm over is equally divided between the two arms of the yoke, the bending moment on each arm is one-half of the twisting moment on the rudder-head.

How far it is possible to diminish the twisting moment on the rudder-head by the use of a *balanced rudder*, is still uncertain; but when one-third of the area of the rudder is afore and the other third abaft its axis of motion, the twisting moment may be approximately estimated at one-third of that exerted on a common rudder of the same dimensions.

The *heel* of the rudder is usually from one-half to two-thirds of the diameter of the head.

A common rudder is supported by being hinged to the stern-post or rudder-post with pintles and braces. According to the ordinary construction of screw-steamers, the rudder hangs from a rudder-post abaft of the screw-opening; and the lower end of that post is connected with the projecting after end of the keel.

There is a bending action, borne partly by the rudder-post and

partly by the rudder itself, whose moment is found approximately as follows:—

V. *Multiply the pressure on the rudder* (as found by Rule I.), *by $\frac{1}{3}$ of the height of the rudder-post, measured from its heel to the counter of the vessel:* for instance, in the example already cited, that moment is $5.4 \times 20 \div 3 = 14$ foot-tons = 168 inch-tons. A *balanced rudder*, which has no rudder-post, but turns on a pivot in the after end of the keel, has to bear the whole of that moment itself; and its moment of resistance may be approximately estimated at being equal to—

VI. *Working modulus of strength \times sectional area of rudder $\times \frac{1}{3}$ of its greatest thickness* (the cross-section of the rudder being treated as an ellipse—see Table under Rule IV. of Article 46 of this Division, Case II.).

The same rule may be applied, without practical error, to the rudder-post, measuring its thickness *diagonally*.

There is also a bending action exerted in a horizontal plane on the projecting after end of the keel, whose moment has the following value:—

VII. *Three-eighths of the pressure on the rudder* (as found by Rule I.) *multiplied by the length of the projecting part of the keel.*

The horizontal moment of resistance of that part of the keel, like that of any other rectangular beam, has the value—

VIII. *Moulding \times (Siding)² \times Working modulus of strength $\div 6$.*

On account of the bending actions just mentioned, it is usual to make the rudder-post and after-end of the keel in screw steamers of twice the sectional area of the main body of the keel, to which the enlarged part gradually tapers.

87. *Strength of Anchors and Cables.*—The details of the construction and fittings of cables and anchors are reserved until the fourth Division of this treatise. The rules by which the weights, dimensions, and testing loads of anchors and cables are regulated in ordinary practice, will be given in the Appendix to this Division. The present Article contains only some general principles as to their strength, with which successful practice is in conformity.

The use of cables and anchors being to hold a ship against forces exerted by the wind and water which tend to make her drift, their strength for different vessels ought to be nearly proportional to the resistances of those vessels at the same speed; and accordingly, the rules followed in ordinary practice lead to the following result; that taking *one-half of the test load* as the working pull upon a bower anchor for a given ship, that working pull is found to be nearly equal to *the resistance of the ship at a speed of twelve knots*. Hence, to find the test load for a ship's bower anchors—

I. *Take double the resistance at 12 knots; or in other words, very nearly, one ton for each 800 feet of augmented surface* (see Division I., Article 162).

Anchors are tested by a load applied at $\frac{1}{3}$ of the length of the arm inwards from the “*pee*,” or point of the bill; and the test load is about $\frac{1}{3}$ of the breaking load; so that the working pull is estimated above with *six* for a factor of safety.

The chain-cable hangs between the ship and the anchor in a curve called a *catenary*. In smooth water the tension at the highest point of that catenary is greater than the tension at the lowest point, by an amount equal to the weight of a piece of the chain itself equal in length to the difference of level of those two points, less the weight of water which such a piece of chain would

displace if hanging vertically,—that is, about $\frac{1}{100}$ of the weight of the iron. A further increase in the tension of the cable is produced by the reaction of part of the chain as it rises and falls with the pitching of the vessel. The chain-cable therefore is liable to a pull greater than that upon the anchor, in a proportion for which no precise rule can be laid down, but for which the provision made in ordinary practice is, that the breaking load of the cable is about *once and a third* that of the anchor.

Chain-cables are also more severely tested than anchors, being subjected to a proof load of at least one-half of their breaking load, and struck with a hammer while under tension; so that—

II. *The test load for a chain-cable is in round numbers about double that of the corresponding anchor, or four times the resistance of the ship at twelve knots; that is to say, one ton for each 400 feet of augmented surface.*

The relations between the *dimensions* of a chain-cable, its test load, and its weight, are determined by the following principles—

The ultimate tenacity of good cable-iron, in the form of straight cylindrical bolts, was found by Sir Samuel Brown (by whom chain-cables were first introduced into practice) to be 750 lbs. *per circular eighth of an inch*; that is, 61,000 lbs. nearly, or 27.3 tons, per square inch.

When welded into oval links, with a cross stay-bar or stud to prevent the sides from collapsing (see Article 20 of this Division, Fig. 1), the tenacity becomes from $\frac{7}{8}$ to $\frac{3}{4}$ of that of the original cable-iron bolts; that is to say, from 24 to 20 tons on the square inch.

The test load adopted is 630 lbs. *on the cable, for each circular eighth of an inch in the area of the original bolts, or 18 tons for each circular inch*; which being reduced to the effective area of the cable itself, is 315 lbs. *per circular eighth of an inch*; or 11.46 tons *on the square inch*. This is expressed more conveniently for practical purposes by the two following rules:—

III. *To find the test load for a given cable, multiply 18 tons by the square of the diameter of the cable-iron in inches.*

IV. *To find the diameter of cable-iron in inches suited to a given test load, divide the load in tons by 18, and extract the square root of the quotient.*

The following are the proportions borne by the dimensions of the links of a chain-cable to the diameter of the bolts of which they are made:—

Length, outside, 6 diameters; inside, 4 diameters.

Breadth, outside, $3\frac{1}{2}$ diameters.

Thickness of stay, at ends, 1 diameter; at middle, $\frac{9}{10}$ diameter.

The following rule gives the *weight of 100 fathoms, in tons, nearly*—

V. *Multiply the square of the diameter, in inches, by 2.43; or otherwise, multiply the test-load by 0.135.*

As the resistance of ships at a given speed is roughly proportional to the square of the cube root of the displacement, the diameter of the iron of their cables, being proportional to the square root of that resistance, is roughly proportional to the cube root of the displacement; and accordingly, Mr. J. R. Napier has proposed the following rule:—

IV. A. *To find the diameter of iron, in inches, for a chain-cable; take $\frac{1}{3}$ of the cube root of the load displacement in tons.*

With regard to the *weight of anchors*, the following rule is given by Mr. J. R. Napier, as closely agreeing with the table sanctioned by the Admiralty:—

VI. Divide the test load of an anchor, in tons, by $2\frac{1}{2}$; the cube root of the fourth power of the quotient will be the weight of the anchor, exclusive of the stock, in cwt., and to the weight thus found $\frac{1}{2}$ may be added as the weight of the stock.

The same author observes, that the weight of an anchor should be proportional to the cube of the diameter of the chain, and gives the following formula as agreeing with practice in the use of Trotman's anchors—

VI. A. Multiply 5 cwt. by the cube of the diameter of the cable-iron in inches.

The results of this rule agree with Lloyd's rules for large vessels, but give somewhat lighter anchors for small vessels.*

Unstudded Chains—that is, chains without the stud or cross-

* For information on the subject of chain-cables, reference may be made to the papers of Mr. G. W. Lenox, in the Transactions of the Institution of Naval Architects for 1860, and of Mr. F. A. Paget, in the Journal of the Society of Arts for the 6th of May, 1864.

stay, which prevents each link from collapsing sideways—are considered to be of about $\frac{2}{3}$ of the strength of *stud-chains*, or chains with stayed links made of the same size of iron.

A *hempen cable* is considered to be of equal strength with a chain-cable, when the *girth* of the hempen cable is ten times the diameter of the iron of the chain-cable.

Each of the *Riding-Bitts* (or upright pieces round which the cable passes on its way from the capstan to the hawse-holes) should be suited to bear the greatest pull on the cable, with a leverage depending on the height of the centre line of the cable where it passes round the bitt, above the centre of the *standard* or horizontal strut, which abuts against the front of the bitt and resists the tendency to pull it forward—say not more than about 30 inches in ordinary cases.

The strength of various other fittings connected with anchors and cables will be considered in the Fourth and Fifth Divisions.

APPENDIX TO THE THIRD DIVISION.

In addition to the authorities on the strength of ships already cited, reference may be made to a paper read by Mr. FAIRBAIRN to the Society of Arts, since the preceding Division was in type.

According to information furnished by Messrs. Jones, Quiggin, & Co., the weight of a steel ship is about *seven-tenths* of that of an iron ship of equal strength. The following are examples:—

Steel Ship " <i>Formby</i> " (see Plates $\frac{P}{1}$ &c.)	
Registered tonnage, 1270.	
Displacement at 20 feet draught,.....	2630 tons.
Weight of cargo,.....	1900 "
Weight of ship, complete,.....	730 "
Weight of iron ship " <i>Seaforth</i> ," of similar model and same size,.....	1013 "
Saving of weight by using steel instead of iron,.....	283 "
Steel Steamer " <i>Hope</i> " (see Plates $\frac{Q}{1}$ &c.)	
Displacement,.....	1308 tons.
Weight of engines and boilers,.....	450
" cargo,.....	338
	788 "
Weight of ship complete,.....	520 "
Estimated weight if built of iron,.....	773 "
Saving by use of steel,.....	253 "

The ultimate tenacity of the steel plates used in building those vessels ranged from 38 to 46 tons per square inch, and averaged about 40 tons per square inch.

RULES LAID DOWN BY UNDERWRITERS.

As examples of good ordinary practice in actual mercantile shipbuilding, the following Tables of Scantlings, together with the figures to which they refer, are extracted (by permission of the respective Committees) from "Lloyd's Register of British and Foreign Shipping," and from the "Underwriters' Registry for Iron Vessels" (Liverpool).

For the details of the Rules which accompany those Tables, reference must be made to the original publications. The following explanations are merely enough to make the tables and illustrations intelligible. It is also to be observed that the Rules are liable to amendment in each year.

With respect to the merit and use of the Rules and Tables of Scantlings it must be held, that as they embody a great collection of facts regarding the structure of ships that have been found to answer in practice, they may be relied upon as insuring at least sufficient strength in all vessels which do not greatly deviate in size, form, and proportions, from the ordinary examples on which the rules were founded.

EXPLANATIONS AS TO LLOYD'S TABLES.

Tables A to F relate to wooden ships; Table G to iron ships; and the Table

called No. 22 to anchors and cables. In every case in which the tonnage of a ship is referred to, the gross tonnage, in tons of 100 cubic feet capacity, is meant, unless otherwise specified (see Division First, Article 99).

WOODEN SHIPS.

Table A relates to the durability of timber, and shows the number of years for which wooden ships, built of the various kinds of timber specified, are to be held entitled to what is called the "Character A," being the highest as to strength and durability. There is another table of durability, containing the same information arranged in a different way, which will be given in the Fourth Division of this treatise.

Tables B and C give the least scantlings of timber for the several parts of wooden ships; and Table D the dimensions of bolts and fastenings. Should the "room and space" be increased beyond that shown in Table B, the siding of the timber is to be increased in the same proportion. Reductions of scantling to the extent of one-fourth are allowed in poops and top-gallant forecastles; the combined length of poop and forecastle not to exceed three-fifths of length of upper deck.

Fig. 1 (Plate $\frac{H}{1}$) shows the arrangement and fastening of *thick strakes*, forming part of the inside planking, and connecting together the floor timbers and first futtocks.

Fig. 2 shows the arrangement of iron diagonal plates or braces required in all wooden ships whose length (measured from the fore part of the stem to the after part of the stern-post on the range of upper deck) exceeds five times the extreme breadth, or eight times the depth. Those braces are to be closely fitted either inside or outside the frames, and are to extend from the upper side of the upper tier of beams to the level of the lower part of chocks amidships, at first futtock heads if fitted outside, and at floor heads if fitted inside. They are to be fastened to each alternate timber if outside, and to each timber if inside, with bolts of sizes not less than those given for "through butt bolts" in Table D. The following are their scantlings and distances apart:—

In ships of 100 tons and under 200 tons,.....	$3\frac{1}{2}$ by $\frac{7}{16}$ inches.
" 200 " " 400 "	4 " $\frac{1}{2}$ "
" 400 " " 700 "	4 " $\frac{5}{8}$ "
" 700 " " 1000 "	$4\frac{1}{2}$ " $\frac{3}{4}$ "
" 1000 " " 1500 "	5 " $\frac{3}{4}$ "
" 1500 and above,.....	$5\frac{1}{2}$ " $\frac{7}{8}$ "

LENGTH OF SHIP.		LEAST DISTANCE.	
Breadth	Depth	Longitudinally.	On the square.
5 to 6	8 to 9	12 feet.	8 feet.
More than 6	9 to 10	10 "	6 "
.....	more than 10	Plans to be submitted to the Committee.	

All such ships to have shelves and water-ways to each tier of beams, each equal in sectional area to the ends of the beams, and bolted through the outside planking at every timber. Ships of lengths exceeding 6 times the breadth or 9 times the depth, to have in addition a rider keelson, or sister keelsons, of sectional area equal to two-thirds of that of the main keelson.

Table C gives the scantlings of beams as fixed by their length amidships. In the after part of the vessel the scantling may be reduced proportionally to the length, but not in the fore part. Iron beams for wooden ships are to be one-eighth of an

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Middle line keelson, I-shaped ($\frac{1}{4}$ Fig. 1), $\frac{3}{8}$ depth of floor-plates; thickness same as that of garboard strakes.

Box keelson (k , Figs. 1, 4), depth, $\frac{3}{8}$ of depth of floor-plates; breadth, $\frac{3}{8}$ of its own depth.

Intercoastal middle line keelson (Figs. 5, 12, 14), web the same thickness with the floor-plates.

Flat-plate keel and intercoastal keelson (Figs. 6, 11, 14).

Centre through-plate keelson (Figs. 1, 2, 7), with flat keelson plate to connect floors; the flat plate to be in thickness equal to the garboard strakes, and $\frac{3}{8}$ of their breadth. Centre through-plate keelson rising above the floors (Figs. 8, 13) to be connected with floors by a pair of flat plates or wings.

Bilge keelsons (see n , Figs. 1, 2).

Intercoastal keelsons, in ships of 1000 tons and upwards, to be midway between middle line keelson and bilge keelsons (o , Figs. 1, 2).

Angle-iron hold stringers, in vessels of 500 tons and upwards (see p , Figs. 1, 2).

Keelsons, and where practicable, stringers also, to be carried through the bulkheads.

The following deductions are allowed from the dimensions in the Table:—

In full poops and top-gallant forecastles, one-fourth. In no case need the outside plating exceed $\frac{1}{8}$ inch in thickness. United length of poop and forecastle not to exceed $\frac{2}{3}$ of length of upper deck.

In parts in the range of upper deck of three-decked ships, except the sheer-strakes, one-sixth.

In parts in range of spar deck of a three-decked ship where depth of hold exceeds $\frac{2}{3}$ of extreme breadth, one-fourth.

Upper and middle deck beams to be fastened to alternate frames.

Depth of Hold, from top of Floor-plates to top of Beams of Second Deck upwards from Hold.		HOLD BEAMS required to be fastened,—
Feet.	Feet.	
12 and under 13 (or gross tonnage above 200),		One to every 8th frame.
13 “ 15,.....		One to every 4th frame.
15 “ 18,.....		{ One to every 2nd and 4th frame alternately.
18 and above,		{ One to every alternate frame; and the same number of middle-deck beams.
ORLOP BEAMS.		
Above 24 feet; and where depth of hold exceeds 15 feet from under side of lower deck beams (in flush-decked ships 1 foot more is allowed),		One to every 6th frame, with stringer plates.

Where spaces between beams exceed two spaces of frames, stringers to be connected with alternate frames by knees or brackets.

Two water-tight bulkheads near the ends are required in steamers; the foremost bulkhead only in sailing vessels. The foremost bulkhead to extend up to the second deck upwards from the hold; the aftermost bulkhead to the same deck, or to a water-tight platform extending above the load-water-line entirely round the after part of the vessel. Vertical angle-iron ribs of bulkheads to be not more than 30 inches apart. The plating of bulkheads to be either rivetted through two frames at each side of the vessel, or, if to one frame, to be stayed longitudinally by brackets or knee-plates to the middle of the side plates, fore side and aft side alternately. Lining pieces in way of bulkheads to be plates extending in one piece from the fore side of the frame afore, to the after side of the frame abaft the bulkhead.

Wood ceiling or lining to be from $1\frac{1}{2}$ inches to 3 inches thick.

The flat of upper deck to be fastened by screw bolts from the upper side, with nuts at the under side of the angle-iron of the beams; where the planks exceed 6 inches in width there must be two bolts in each plank in every beam, one of which may be a short screw bolt, provided the planks do not exceed 8 inches in width, in which case both bolts must be put through. The water-ways, if of wood, to be fastened with screw bolts with nuts at under side of stringer plates.

ANCHORS AND CABLES.

Anchors and cables to be tested at a public machine. For number, weight, and test-loads, see Table, No. 22. When hempen cables are used, one-sixth more length is required than for chain cables.

EXPLANATION AS TO THE LIVERPOOL TABLES OF SCANTLING FOR NEW VESSELS.

The rules relating to those tables are in this work printed on the sheet of tables in form of notes. Some of the rules and tables relate to spars and rigging. The Table of Anchors, Cables, &c., is, with the exception of two lines and a column, the same with the Table No. 22 of Lloyd's Rules, to which reference is therefore made, instead of repeating it.

No. 22.]

ANCHORS AND CABLES.

Minimum Weights of Anchors of unobjectionable form and proportions; Sizes and Lengths of Chain Cables, and the proof strain to which they are to be tested; and Sizes and Lengths of Hawser and Warps.

SHIP'S TONNAGE.	ANCHORS.									STUD-CHAIN CABLES.†			HAWERS AND WARPS.					SHIP'S TONNAGE.
	Number.			Weight.						Minimum Size.	Proved to Admiralty Test.	Length.	Stream.		Hawser.	Warp.	Length.	
													Chain.	Rope.				
	Bowers.*	Stream.	Kedges.	Ex. Stock.	Admiralty Test.	Stream.	Kedge.	2nd Kedge.										
Tons.				Cwts.	Tons.	Cwts.	Cwts.	Cwts.	Inches†	Tons.	Fathoms.		Inch.	Inch.	Inch.		Tons.	
50	2	1	1	2½	4½	1	½	—	1½	8½	120		5	3	—	—	50	
75	2	1	1	2¾	5½	1½	¾	—	1¾	10½	120		5	3	—	—	75	
100	2	1	1	4	6½	1¾	1	—	1¾	11½	150		5½	3	—	—	100	
125	2	1	1	5½	7½	2	1	—	1¾	13½	180		5½	3½	—	—	125	
150	2	1	1	6	8½	2½	1½	—	1¾	15½	180		6	4	—	—	150	
175	2	1	1	7½	9½	3	1¾	—	1	18	180		6	4	—	—	175	
200	3	1	1	8½	10½	3	1¾	—	1½	20½	180		6½	4	—	—	200	
250	3	1	2	10	12	4½	2½	1	1½	22½	210		7	5	—	—	250	
300	3	1	2	12	13½	5	2½	1½	1½	25½	210		7½	5½	—	—	300	
350	3	1	2	13½	15½	6	3	1½	1½	28½	240		7½	5½	—	—	350	
400	3	1	2	15½	16½	6½	3½	1½	1½	31	240		8	6	—	—	400	
450	3	1	2	16½	18	7	3½	1½	1½	34	270		8½	6½	—	—	450	
500	3	1	2	18	19	8	4	2	1½	37½	270		9	7	—	—	500	
600	3	1	2	21	21½	9	4½	2½	1½	40½	270		9½	7	4	—	600	
700	3	1	2	23½	23½	10	5	2½	1½	44	300		10	8	5	—	700	
800	3	1	2	25½	25½	10½	5½	2½	1½	47½	300		10	8	5	—	800	
900	3	1	2	27½	26½	11	5½	2½	1½	51½	300		10	9	5½	—	900	
1000	3	1	2	30	28½	12	6	3	1½	55½	300		10	9	5½	—	1000	
1200	3	1	2	32	30½	13	6½	3½	1½	59½	300	1	10	9½	6	—	1200	
1400	3	1	2	34	31½	13½	6¾	3½	1½	63½	300	1	10	10	6	—	1400	
1600	3	1	2	36½	33½	14	7	3½	1½	67½	300	1	11	10½	6½	—	1600	
1800	3	1	2	38	34½	14½	7½	3½	2	72	300	1	11	11	7	—	1800	
2000	4	1	2	40	35½	15	7½	3½	2½	76½	300	1	11	11	7	—	2000	
2500	4	1	2	42	37½	17	8½	4½	2½	81½	330	1	12	12	8	—	2500	
3000	4	1	2	45	39½	19	9½	4½	2½	91½	360	1	12	12	8	—	3000	

MEM.—For Steamers the Anchors and Cables will not be required to exceed in weight and length those of a sailing vessel of two-thirds their total tonnage.

* Two of the Bower Anchors must not be less than the weight set forth above, but in the third a reduction of 15 per cent. will be allowed.—All Anchor Stocks must be of acknowledged and approved description.

† Unstudded close-link Chains of 1 inch in diameter and under, will be admitted as Cables, if proved to two-thirds the test required for stud-chains. But in all such cases a short length, not less than twelve links, must be tested up to the full strain for stud-link chains.

‡ Parties desirous of using or supplying chains of smaller size, and who are willing to subject them to a greater strain than set forth above, may submit their propositions to the Committee.

TABLE A.—(FROM LLOYD'S RULES.)

DURABILITY OF TIMBER.—EXHIBITING THE NUMBER OF YEARS TO BE ASSIGNED TO THE DIFFERENT DESCRIPTIONS OF TIMBER USED IN SHIPS, THE SAME TO BE OF GOOD QUALITY, PROPERLY SEASONED, AND FREE FROM DEFECTS.

	TIMBERING.										OUTSIDE PLANK.					INSIDE PLANK, &c.		
	Floors.	First Futlocks.	Second Futlocks.	Third Futlocks and Top Timbers.	Main and Rider Keelsons.	Stem and Stern Post.	Transoms, Knight-heads, Hawse-Timbers, Apron, and Deadwood.*	Beams and Holes.	Knees.	Rudder and Windlass. — Main Pieces.	Keel to First Futlock Heads.	First Futlock Heads to Light Mark.	Light Mark to Wales.	Wales, Black-Strakes, Top-sides, and Sides.	Upper Deck Water-way, Spiketting, and Plank-sheer.	Shelves, Clamps, Limber and Bilge Strakes, Ceiling in Hold and between Decks, also Spiketting and Water-way below the Upper Deck.		
1	English, African, and Live Oak, Adriatic, Italian, Spanish, Portuguese, and French Oak; East India Teak, Morung Saul, Greenheart, Morra, Iron Bark, Chow, Red Kranje, Kapur,....	12	12	12	12	12	12	12	12	12	12	12	12	12	12	English, African, and Live Oak, Adriatic, Italian, Spanish, Portuguese, and French Oak; East India Teak, Morung Saul, Greenheart, Morra, and Iron Bark.		
2	Mahogany of hard texture, Cuba Sabicu, Pencil Cedar, Angelly, and Venatic.	10	10	10	10	10	10	12	12	10	12	12	10	10	10	Mahogany of hard texture, Cuba Sabicu, Pencil Cedar, Angelly, and Venatic.		
3	Other Continental White Oak, Spanish Chestnut, and Blue Gum,.....	9	9‡	7	7	9	7	8	8	7	12	12	9	8	9	Other Continental White Oak, Spanish Chestnut, and Blue Gum.		
4	N. American White Oak, American Sweet Chestnut, Stringy Bark, and Red Cedar.	8	8‡	7	7	8	7	7	7	7	12	10	8	7	7	N. American White Oak, American Sweet Chestnut, Stringy Bark, and Red Cedar.		
5	Pitch Pine, Larch, Hackmatack, Tamarac, and Juniper,.....	7	7	7	7	8	7	8	8	7	12	10	8	8	10	Pitch Pine, Larch, Hackmatack, Tamarac, and Juniper.		
6	Second-hand English Oak, African Oak, and East India Teak,§§.....	7	7	6	6	6	5	6	6	5	—	—	—	—	5	Second-hand English Oak, African Oak, and East India Teak,§§		
7	Cowdie, Huon Pine,.....	6¶	6	6	7	7	6	6	7	—	10	9	8	7	10	Cowdie, Huon Pine.		
8	Baltic and American Red Pine,.....	5	5	5	7	7	5	5	7	5	9	9	8	7	10	Baltic and American Red Pine.		
9	English Ash,.....	5	5	5	5	5	4	4	5	5	10	7	4	—	—	English Ash.		
10	Foreign Ash,.....	5	5	4	4	5	4	4	5	5	—	10	7	4	—	Foreign Ash.		
11	American Rock Elm and Hickory,.....	6¶	6	5	5	6	5	5	5	4	12§	8	6	5	5	American Rock Elm and Hickory.		
12	European and American Grey Elm,.....	5	5	4	4	4	4	4	5	5	—	12§	8	5	4	European and American Grey Elm.		
13	Black Birch and Black Walnut,.....	5¶	5**	4	4	4	4	4	4	4	10	7	4	4	4	Black Birch and Black Walnut.		
14	Spruce Fir and White Cedar,††.....	5	5**	4	4	4	4	4	4	7	6	6	5	4	4	Spruce Fir and White Cedar.††		
15	Beech,.....	5¶	4	—	—	4	—	—	—	4	12§	8	4	—	—	Beech.		
16	Yellow Pine,.....	—	—	—	4	4	4	4	4	—	6	5	5	5	5††	Yellow Pine.		
17	Hemlock,.....	4	4	4	4	—	—	—	4	—	4	4	4	4	4	Hemlock.		

* This Table applies as to the Deadwood so far as regards the Material to be used from the height of two feet above the rabbet of the Keel.
† American Rock Elm allowed for Limber Strakes, and Ceiling between them in Ships of the 7 years' grade.
‡ If the First Futlocks run up above the Light Watermark, the use of Foreign White Oak is allowed for the 7 years' grade only.
§ The use of Elm and Beech, in Ships above the 8 years' grade, to be restricted to a height from the lower part of the Main Keel, of one-third of the internal depth of the Ship measured, in midships, from the top of the Limber Strake to the top of the Upper Deck Beams.
|| The Materials marked thus † under the head of "Rudder and Windlass," allowed in Ships of 300 tons and under only.
¶ Black Birch, Beech, American Rock Elm, and Cowdie, allowed for Floors in Midships, to an extent not exceeding one-half the entire length of the Keel in Ships of the 7 years' grade.
** Black Birch and Spruce allowed for First Futlocks amidships, to the same extent in Ships of the 6 years' grade.
†† Yellow Pine allowed for Waterways of Upper Deck in Ships of the 7 years' grade, if properly fastened, as prescribed in Table B, and provided the Beams are well secured independently of the Waterways.

‡‡ White Cedar allowed for Third Futlocks and Top timbers in Ships of the 7 years' grade.
§§ In cases where second-hand Teak of approved quality is proposed to be used, application may be made to the Committee with a view to it being allowed a higher grade (not exceeding two years) than as set forth above.
M.M.—The word "English" includes Timber the growth of the United Kingdom.
MALAYAN TIMBER.—Ballow wood will be allowed for all parts in Ships of the highest grade.
Sama, Samaram, and Minick Kruen.—These woods will be admitted to an equality in classification with the Timbers named in line 2 of Table A.
Santa Maria Wood will be allowed for any part in Ships of 9 years' grade; for beams in Ships of 10 years' grade; and for outside planking to light water mark in Ships of 12 years' grade.
All the above-named tropical woods must be of good and unexceptionable quality to entitle them to classification as above; and, as they are as yet but little known in this country, they will not, at present, be entered by name in Table A.

TABLE B.—(FROM LLOYD'S RULES.)

SCANTLINGS OF WOODEN SHIPS.—MINIMUM DIMENSIONS OF TIMBERS, KEELSON, KEEL, PLANKING, &c.

TONNAGE (GROSS),.....Tons.																					For Moulding, see Foot-note.
	50	100	150	200	250	300	350	400	450	500	550	600	650	700	750	800	850	900	950	1050	
* TIMBER AND SPACE,.....INCHES	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36	37	
Floors, sided and moulded at Keelson, if squared,.....	7	7	8	8	9	10	11	11	12	13	13	13	13	13	13	13	13	13	13	13	
Double Floors, sided and moulded at Keelson, if squared,.....	6	6	7	7	8	9	10	10	11	12	12	12	12	12	12	12	12	12	12	12	
† 1st Futlocks, sided and moulded at Floor Heads, if squared,.....	6	6	7	7	8	9	10	10	11	12	12	12	12	12	12	12	12	12	12	12	
2nd Futlocks, sided, if squared,.....	5	5	6	6	7	8	9	9	10	11	11	11	11	11	11	11	11	11	11	11	
3rd Futlocks and Long Top Timbers, sided, if squared,.....	5	5	6	6	7	8	9	9	10	11	11	11	11	11	11	11	11	11	11	11	
Top Timbers (Short), sided, if squared,.....	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	
Top Timbers, moulded at heads, if squared,.....	4	4	4	5	5	5	5	5	6	6	6	6	6	6	6	6	6	6	6	6	
Breasthooks & Wing Transom, sided & moulded in the middle,.....	8	8	9	9	10	10	11	11	12	13	13	13	13	13	13	13	13	13	13	13	
† Keel, Stem, Apron, and Sternpost, sided and moulded,.....	8	9	10	10	11	11	12	12	13	14	14	14	14	14	14	14	14	14	14	14	
Keelson, also the Mainpiece of Rudder from lower part of Counter upwards, sided and moulded,.....	9	10	11	11	12	12	13	13	14	15	15	15	15	15	15	15	15	15	15	15	
§ Wales,.....	3	3	4	4	4	4	4	4	5	5	5	5	5	5	5	5	5	5	5	5	
Bottom Plank, from Keel to Wales,.....	2	2	2	2	2	3	3	3	3	4	4	4	4	4	4	4	4	4	4	4	
Sheer Strakes, Top-sides, Upper Deck Clamp where there is no Shelf fitted, and Lower Deck Clamp with a Shelf,.....	2	2	3	3	3	3	3	3	4	4	4	4	4	4	4	4	4	4	4	4	
Ceiling below Hold Beam Clamp,.....	1	1	2	2	2	2	2	2	3	3	3	3	3	3	3	3	3	3	3	3	
¶ Waterway,.....	3	4	4	5	5	5	5	6	6	6	6	6	6	6	6	6	6	6	6	6	
Ceiling between Decks,.....	1	1	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2	
Bilge Plank, inside, Thick Strakes over long and short,.....	2	2	3	3	3	3	3	3	4	4	4	4	4	4	4	4	4	4	4	4	
Floor Heads, and Limber Strake,.....	2	2	3	3	3	3	3	3	4	4	4	4	4	4	4	4	4	4	4	4	
Lower Deck Clamp where there is no Shelf fitted, and Spiketting,.....	—	—	3	3	3	3	4	4	4	4	4	4	4	4	4	4	4	4	4	4	
Upper Deck Clamp where a Shelf is also fitted,.....	2	2	2	2	2	2	2	2	3	3	3	3	3	3	3	3	3	3	3	3	
Plank-sheer,.....	2	2	2	2	2	2	2	2	3	3	3	3	3	3	3	3	3	3	3	3	
Flat of Upper Deck,.....	2	2	2	2	2	2	2	2	3	3	3	3	3	3	3	3	3	3	3	3	
Scarp of Keelson without Rider,.....	4	4	4	5	5	5	5	5	6	6	6	6	6	6	6	6	6	6	6	6	
Ditto, where Rider Keelson is added, also Scarps of Keel,.....	4	4	4	5	5	5	5	5	6	6	6	6	6	6	6	6	6	6	6	6	

Moulding of Futlocks and Top Timbers to diminish gradually from size given at Floor Heads to that at Top Timber Heads.
* Should the timber and space be increased, the siding of the timbers to be increased in proportion.
† When the heels of 1st Futlocks meet at the middle line on the Keel, under the Keelson, either with full moulding, or with Cross Chocks properly battled, the siding of single Floors, and their moulding at the Keelson, may be reduced to the siding and moulding allowed for Double Floors.
‡ The rabbet of the Keel, Stem, and Sternpost to be made so as to leave sufficient substance of wood to form a substantial back rabbet.
§ For Breadth of Wales required in every case, see Rules.

|| All the fore and after heads, both outside and inside, may be reduced one-sixth in thickness. Furrows are not allowed in this or in any other part of a ship.
¶ This Depth of Waterway for Frying Surface against Timbers is required, below the underside of the Plank-sheer, to receive in and out through Bolts at alternate Timbers, with alternate through bolts in Shelf, and in Clamp where there is no Shelf.
M.M.—For relaxations in respect to Poops, Top-gallant forecastles, and raised quarter decks, see Rules. For requirements for vessels of excessive length as compared with breadth and depth, see Rules.

TABLE C.

SIDING AND MOULDING OF BEAMS.

LENGTH OF BEAM amidships.	HOLD BEAMS.		DECK BEAMS.	
	Sided and moulded at ends.	Moulded at ends.	Sided and moulded at ends.	Moulded at ends.
10	—	—	4	4
11	—	—	5	4
12	—	—	5	4
13	—	—	5	4
14	—	—	5	4
15	8	6	6	5
16	8	7	6	5
17	8	7	6	5
18	9	7	7	5
19	9	8	7	6
20	10	8	7	6
21	10	8	7	6
22	10	9	8	6
23	11	9	8	6
24	11	9	8	7
25	11	9	8	7
26	12	10	8	7
27	12	10	9	7
28	12	10	9	7
29	12	10	9	7
30	13	11	9	8
31	13	11	9	8
32	13	11	9	8
33	13	11	10	8
34	14	11	10	8
35	14	12	10	8
36	14	12	10	8
37	15	12	10	8
38	15	12	10	8
39	15	12	10	9
40	15	13	10	9

N.B.—The size of Orlop Beams to be the mean of the sizes here prescribed.

TABLE D.—(FROM LLOYD'S RULES.)

WOODEN SHIPS.—SIZES OF BOLTS, PINILES OF RUDDER, AND TREENAILS.

TONNAGE (GROSS),.....Not exceeding Tons	50	100	150	200	250	300	350	400	450	500	700	900	1350
Heel-Knee, Stemson, and Deadwood Bolts,.....Inches	1	1	1	1	1	1	1	1	1	1	1	1	1
Bolts in Sister Keelsons, Scarps of Keel,* Arms of Breasthooks, Pointers, Crutches, Riders, Hanging and Lodging Knees to Hold or Lower Deck Beams (except in and out Throat Bolts of Hanging Knees, which must be larger), also in and out Bolts of Shelf, Clamp, and Waterway of Hold or Lower Deck Beams, and the in and out Throat Bolts of Upper Deck Hanging Knees,.....	1	1	1	1	1	1	1	1	1	1	1	1	1
Keelson Bolts (one through Keel at each Floor), Throats of Transoms, Throats of Breasthooks, and throats of Hanging Knees to Hold or Lower Deck Beams,.....	1	1	1	1	1	1	1	1	1	1	1	1	1
Bilge, Limber Strake, and through Butt Bolts,.....	1	1	1	1	1	1	1	1	1	1	1	1	1
Other Butt Bolts,.....	1	1	1	1	1	1	1	1	1	1	1	1	1
Bolts through heels of cant timbers at fore and after Deadwood. In and out bolts of Upper Deck Waterway, Shelf and Clamp, also Arms of Hanging and Lodging Knees, except in and out Throat Bolts of Hanging Knees, which must be larger,.....	1	1	1	1	1	1	1	1	1	1	1	1	1
Piniles of Rudder { The Braces of which must extend so as to receive not less than Two Bolts on the Planking on each side,.....	2	2	2	2	2	2	2	2	2	2	2	2	2
Hardwood Treennails,.....	1	1	1	1	1	1	1	1	1	1	1	1	1

* NUMBER OF BOLTS IN SCARPS OF KEEL.—In Ships of 150 tons and under, 6 bolts; above 150 tons and under 500 tons, 7 bolts; 500 tons and above, 8 bolts.
N.B.—Bolts to be through and clenched, as prescribed in the Rules.

TABLE E.

NUMBER OF HANGING KNEES.

Tons.	To Hold Beams.	To Upper Deck Beams.
150	4	4
200	4	6
250	5	7
300	6	8
350	7	9
400	8	10
450	8	11
500	9	12
550	9	13
600	10	14
650	10	15
700	11	16
750	11	17
800	12	18
900	13	20
1000	14	22
1100	15	24
1350	17	26

TABLE F.—(FROM LLOYD'S RULES.)

WOODEN SHIPS.—MINIMUM DIMENSIONS OF IRON KNEES AND KNEE RIDERS FOR BRITISH NORTH AMERICAN BUILT SHIPS AND FIR SHIPS.

TONNAGE,.....Tons	150	200	250	300	350	400	450	500	550	600	650	700	750	800	900	1000	1100	1200	1300	1400	1500	1600	1700	1800	1900	2000
Number of Hanging Knees to Hold or Lower Deck Beams,.....Pairs)	3*	4	6	8	9	Upwards, one Knee Rider to every Beam, or Knees and Riders as per Section 62.																				
Number of Hanging Knees to Upper and Middle Deck Beams,.....Pairs)	4	6	7	8	9	10	11	12	13	14	15	16	17	18	Upwards, one to every Beam.											
Breadth of Knees and Riders to Hold or Lower Deck Beams,.....Inches)	8	8	8	8	8	8	8½	8½	8½	8½	8½	8½	4	4	4½	4½	4½	4½	4½	4½	5	5	5½	5½	5½	5½
Breadth of Upper Deck Knees, where there are two Decks, and of Middle Deck Knees, where there are three Decks,.....Inches)	3	3	3	3	3	3	3½	3½	3½	3½	3½	3½	3½	3½	4	4	4½	4½	4½	4½	4½	4½	4½	4½	4½	4½
Thickness of Riders at the joints or butts of the Timbers,.....Inches)	1½	1½	1½	1½	1½	1½	1½	1½	2	2	2½	2½	2½	2½	2½	2½	3	3	3½	3½	3½	3½	3½	3½	3½	3½
Thickness of Knees to Lower Deck or Hold Beams and Knee Riders at the Angle of the Throat,.....Inches)	2½	2½	2½	2½	3	3	3½	3½	3½	3½	3½	3½	4	4	4½	4½	4½	4½	4½	4½	5	5	5½	5½	5½	5½
Thickness of Knees to Lower Deck or Hold Beams and Knee Riders at the Throat Bolts,.....Inches)	1½	1½	2	2	2½	2½	2½	2½	2½	2½	2½	2½	3	3	3	3	3½	3½	3½	3½	3½	3½	3½	3½	3½	3½
Thickness of Knees to Upper or Middle Deck at the Throat Bolts,.....Inches)	1½	1½	1½	1½	2	2	2½	2½	2½	2½	2½	2½	2½	2½	2½	2½	3	3	3	3	3½	3½	3½	3½	3½	3½
Thickness of Hanging Knees (not Riders) at the ends,.....Inches)	8	8	8	8	8	8	8	8	8	8	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1
Length of Beam Arms of Knees and Knee Riders for Lower Deck or Hold Beams,.....	ft. 2 in. 6	ft. 2 in. 6	ft. 2 in. 9	ft. 2 in. 9	ft. 3 in. 0	ft. 3 in. 0	ft. 3 in. 3	ft. 3 in. 3	ft. 3 in. 3	ft. 3 in. 3	ft. 3 in. 6	ft. 3 in. 6	ft. 3 in. 6	ft. 3 in. 9	ft. 3 in. 9	ft. 3 in. 9	ft. 3 in. 9	ft. 4 in. 0	ft. 4 in. 0	ft. 4 in. 0	ft. 4 in. 0	ft. 4 in. 0	ft. 4 in. 0	ft. 4 in. 0	ft. 4 in. 0	ft. 4 in. 0

NOTE.—The Bolts in all Iron Riders in Hold, to be not more than twenty-one inches apart on the average. Standards upon the Beams of such Ships are not admitted as substitutes for Hanging Knees below them. For sizes of Bolts, see Table D.
 * Provided the depth of hold be 13 feet or upwards.
 † Breadth and thickness of Knees for Upper Deck, where there are Three Decks, may be one-sixth less.
 ‡ Beam Arms of Upper and Middle Deck Knees, may be three inches shorter than those of the Lower Deck. Side Arms of Hanging Knees not to be less in length than one and a half the length of their Beam Arms. Beam Arms of Knees and Knee Riders, which are 8 feet 6 inches in length, to have not less than Four Bolts; and shorter than that length, to have not less than Three Bolts. Side Arms of all Hanging Knees to have at least One Bolt more than in the Beam Arms.

TABLE G.—(FROM LLOYD'S RULES.)

IRON SHIPS.—TABLE OF MINIMUM DIMENSIONS OF FRAMES, PLATING, RIVETS, KEELS, KEELSONS, STEMS, STERN POSTS, FLOOR PLATES, BEAMS,‡ BULKHEADS, STRINGERS, &c.

All plates, and all beam and angle iron, used in ships intended for classification, are to be stamped legibly in two places with the manufacturer's trade mark, or his name and the place where made.

GROSS TONNAGE.	Keel, Stem, and Stern Post for all Grades.*	Distance of Frames from Moulding edge to Moulding edge all fore and aft for all Grades.	FRAMES. Dimensions of Angle Iron for all Grades.	Dimensions of Reversed Angle Iron on Frames, Bulkheads, and Box Keelsons, for all Grades.	THICKNESS OF OUTSIDE PLATES.†								Thickness of Stringer Plates upon Beams, Floor Plates, Hooks, Crutches, and Box, or Interstitial Keelsons for all Grades.	Thickness of Plates for Bulkheads for all Grades.	Dimensions of Angle Iron on Beam Stringers or Keelsons for all Grades.	RUDDER for all Grades.		Thickness of Wood Flat of Upper Deck.	GROSS TONNAGE.						
					Garboard Strakes* and Single Plate Middle Line Keelsons standing upon Floors.											From the Garboard to the upper part of Bilge, and the Sheerstrakes.*				From upper part of Bilge to a perpendicular height from upper side of Keel of three- fifths the internal depth of the upper side of upper Hold, measured from the upper part of Bilge, and the Sheerstrakes.‡		From three-fifths the depth of Hold (measured from the upper side of upper deck in all Ships, whether spar deck or otherwise) to lower edge of Sheerstrake.		Diameter at the Head.	Diameter at the Heel.
					⏏	⏏	⏏	⏏	⏏	⏏	⏏	⏏				⏏	⏏			⏏	⏏	⏏	⏏	⏏	⏏
100 and under 200	Inches. 6 × 1½	If single frames be adopted, the space from centre to centre is not to exceed 21 inches, all fore and aft; but provided an additional frame, for half the vessels length amidships, be fitted at opposite side of each floor plate across the keel, and extended to upper part of bilges, and rivetted through floor plates and main frames, also through the outside plating, as required for main frames, the space may be increased to 23 inches in ships under 1000 tons, and to 24 inches in ships of 1000 tons and upwards.	8/16 × 2½ × 2½	8/16 × 2½ × 2½	8/16	7/16	7/16	7/16	6/16	6/16	6/16	6/16	8/16	8/16	8/16 × 3 × 3	3	2	2½	{ 100 and under 200						
200 and under 300	6½ × 2		8/16 × 3 × 2½	8/16 × 2½ × 2½	8/16	7/16	7/16	7/16	7/16	7/16	7/16	7/16	8/16	8/16	8/16 × 3 × 3	3½	2	2½	{ 200 and under 300						
300 and under 400	6½ × 2½		8/16 × 3½ × 2½	8/16 × 2½ × 2½	10/16	9/16	9/16	9/16	8/16	7/16	7/16	7/16	8/16	8/16	8/16 × 3½ × 3	3¾	2½	3	{ 300 and under 400						
400 and under 500	6½ × 2½		7/16 × 3½ × 2½	7/16 × 2½ × 2½	10/16	9/16	9/16	9/16	8/16	7/16	7/16	7/16	8/16	8/16	8/16 × 4 × 3	4½	2½	3	{ 400 and under 500						
500 and under 600	7 × 2½		7/16 × 3½ × 2½	8/16 × 3 × 2½	11/16	10/16	10/16	10/16	9/16	8/16	8/16	8/16	9/16	9/16	7/16 × 4½ × 3½	4½	2½	3½	{ 500 and under 600						
600 and under 700	7 × 2½		7/16 × 4 × 3	8/16 × 3 × 2½	11/16	10/16	10/16	10/16	9/16	8/16	8/16	8/16	9/16	9/16	7/16 × 4½ × 3½	4½	2½	3½	{ 600 and under 700						
700 and under 800	7½ × 2½		8/16 × 4½ × 3	7/16 × 3 × 2½	11/16	11/16	11/16	11/16	10/16	9/16	9/16	9/16	10/16	10/16	8/16 × 4½ × 3½	5	3	3½	{ 700 and under 800						
800 and under 900	7½ × 3		8/16 × 4½ × 3	7/16 × 3 × 3	11/16	11/16	11/16	11/16	10/16	9/16	9/16	9/16	10/16	10/16	8/16 × 5 × 4	5½	3	3½	{ 800 and under 900						
900 and under 1000	8 × 3		9/16 × 4½ × 3	7/16 × 3½ × 3	11/16	11/16	11/16	11/16	11/16	10/16	10/16	10/16	11/16	11/16	9/16 × 5 × 4½	5½	3	3½	{ 900 and under 1000						
1000 and under 1200	8½ × 3		9/16 × 5 × 3	8/16 × 3½ × 3	11/16	11/16	11/16	11/16	11/16	10/16	10/16	10/16	11/16	11/16	9/16 × 5 × 4½	5¾	3	4	{ 1000 and under 1200						
1200 and under 1500	9 × 3		9/16 × 5 × 3½	8/16 × 3½ × 3	11/16	11/16	11/16	11/16	11/16	11/16	11/16	11/16	11/16	11/16	9/16 × 5½ × 4½	6	3½	4	{ 1200 and under 1500						
1500 and under 2000	10 × 3		10/16 × 5½ × 3½	9/16 × 4 × 3½	11/16	11/16	11/16	11/16	11/16	11/16	11/16	11/16	11/16	11/16	9/16 × 6 × 5	6½	3½	4	{ 1500 and under 2000						
2000 and under 2500	12 × 3		10/16 × 6 × 4	9/16 × 4½ × 3½	11/16	11/16	11/16	11/16	11/16	11/16	11/16	11/16	11/16	11/16	10/16 × 6½ × 5½	7½	3½	4	{ 2000 and under 2500						
2500 and under 3000	12 × 3½		11/16 × 6½ × 4	10/16 × 4½ × 3½	11/16	11/16	11/16	11/16	11/16	11/16	11/16	11/16	11/16	11/16	10/16 × 6½ × 5½	7¾	4	4	{ 2500 and under 3000						
3000 and under 3500	12 × 3½		11/16 × 6½ × 4	10/16 × 4½ × 3½	11/16	11/16	11/16	11/16	11/16	11/16	11/16	11/16	11/16	11/16	10/16 × 6½ × 5½	8	4½	4	{ 3000 and under 3500						

MEM.—The scantlings given in the above Table are intended for Ships the length of which, measured from the fore part of the Stem to the after part of the Stern-post on the range of the Upper Deck, does not exceed seven times their extreme breadth or ten times their depth of Hold, taken from the upper part of Floors to the top of the Upper Deck Beams.

RIVETS. Diameter of Rivets required for Thickness of Plates, .	5/16 of an inch.			7/16 of an inch.			1 of an inch.			1 1/8 of an inch.		
	1/16	1/8	3/16	1/8	3/16	1/4	1/4	3/8	1/2	5/8	3/4	7/8
	1/16	1/8	3/16	1/8	3/16	1/4	1/4	3/8	1/2	5/8	3/4	7/8

Rivets to be ½ of an inch larger in diameter in the stem, stern-post, and keel.

* Hollow or flat keel plates (vide Engravings, Figs. 6, 11, and 14) and garboard strakes, and main sheerstrakes, are not to be less in breadth than as follows, viz. :—In ships under 500 tons, 2ft.; in ships 500 and under 1000 tons, 2ft. 6in.; in ships 1000 tons and upwards, 3ft. When Hollow or Flat Plate Keels are adopted, their thickness should not be less than one and a half that of the Garboard Strake. For Keels of other Forms, see Engraving, Figs. 7, 8, 9, 10, and 13.

† PLATING.—No plates to be less in length than five spaces of frames (vide Fig. 2), except the Fore and after hoods. No butts of outside plating in adopted strakes to be nearer each other than two spaces of frames (vide Fig. 2). In vessels under 1200 tons the plating may be reduced from the thickness shown in Table, one-sixteenth of an inch forward and aft, for a distance not exceeding one-quarter of the length of the vessel from each end, below the upper edge of main sheerstrake, down to a perpendicular height from upper side of keel of three-fifths the internal depth of hold, including the height of the spar deck in spar-deck ships; and in ships of 1200 tons and upwards, a reduction of two-sixteenths will be allowed; the plates next abaft and next afore the quarter length to be of an intermediate or graduated thickness, between that required amidships and the reduction allowed at the ends. In screw-propelled vessels, however, no reduction is to be made in the plating at the after end, below the lower part of the rudder trunk.

BUTT STRAPS.—All plates are to be well fitted, and secured to the frames and to each other; the butts to be closely fitted by planing or otherwise, and to be united by butt straps, of not less than the same thickness as the plates, and of sufficient breadth for rivetting, as described hereafter, and to be fitted with the fibre of the iron in the same direction as the fibre of the plates to which they are rivetted; the space between the plating and the frames to have solid filling or lining pieces, closely fitted in one length, and of the same breadth as the frames. It is recommended that in all cases the sheerstrake be an outside strake, so as to admit of the butt straps or lining pieces being extended, in one piece, from the fore side of the frame next afore the butts to the aft side of the frame next abaft the butts (vide g, Fig. 2), or to admit of doubling the sheerstrake where it may be required. For breadth of sheerstrake see footnote above.

‡ BEAMS.—Beam plates to be in depth one-quarter of an inch for every foot in length of the midship beams, and to be in thickness one-sixteenth of an inch for every inch in depth of the said beams, and to be made of T bulb iron, or bulb plate with double angle irons rivetted on upper edge; the two sides of each of these angle irons to be not less in breadth than three-fourths the depth of beam plate, and to be in thickness one-sixteenth of an inch for every inch of the two sides of the angle iron; where beams below the upper or middle deck (including orlop beams) have no deck laid upon them, the angle irons on their upper edges to be of the dimensions required for the angle iron of the reverse frame; or the beams may be composed of any other approved form of beam iron, equal to the strength herein required. All beams to be well and efficiently connected or rivetted to the corresponding frames, at the sides of the vessel, with bracket ends or knee plates; each arm of knee plates at ends of beams not to be less in length than twice and half the depth of beams, and to be in thickness equal to the beams. The beams of each deck to be placed over each other, and pillared where practicable.

RIVETS AND RIVETTING.—The Rivets to be of the best quality, and to be in diameter as per Table; the rivet holes to be regularly and equally spaced and carefully punched opposite each other from the faying surfaces, in the laps and lining pieces or butt straps; and to be counterbored all through the outer plating (vide Fig. 16); the rivets not to be nearer to the butts or edges of the plating, lining pieces to butts, or of any angle iron, than a space not less than their own diameter, and not to be further apart from each other than four times their diameter, or nearer than three times their diameter, and to be spaced between the beams and outside plating, and in reversed angle iron, a distance equal to eight times their diameter apart. When rivetted up they are completely to fill the holes, and their points or outer ends are to be round or convex (vide Fig. 16), and not to be below the surface of the plating through which they are rivetted. All vessels to have all edges or horizontal joints of outside plating double rivetted from the keel to the height of upper part of bilges (to d, Fig. 1), all fore and aft; but vessels of 700 tons and above,

intended for the highest grade, are to have all edges or horizontal joints of outside plating double rivetted throughout (vide Fig. 2). The stem, stern-post, keel, edges of garboard strakes and sheerstrakes, and butts of outside plating, and butts of floor plates, transoms, and plates of beams, also butts of keelsons, stringers, shelf plates, and all longitudinal ties, to be double rivetted in all vessels. The overlaps of plating, where transverse rivetting is required, not to be less than five and a half times the diameter of the rivets (vide Fig. 16); and where single rivetting is admitted, to be not less in breadth than three and a quarter times the diameter of the rivets. If double rivetting be adopted where single rivetting is allowed by the Rules, the diameter of the rivets may be reduced one-sixteenth of an inch below that prescribed by the Rules, provided that in no case the diameter be less than five-eighths of an inch. The butts and edges of outside plating to be truly fitted, carefully caulked, and made water-tight.

§ FLOOR PLATES.—The floor plates to be in depth at middle line according to the following rule, viz. :—To the vessel's depth, measured from the top of keel to the top of upper or spar deck beams amidships, add the extreme breadth of the vessel; two-fifths of that sum in inches, to be the depth of the floor plates at middle line; their thickness to be as given in Table; but at each end of the vessel, for one-quarter of her length, they may be reduced in thickness one-sixteenth of an inch where the plates are less than ten-sixteenths, and two-sixteenths of an inch where the plates are ten-sixteenths and upwards. The floor plates to extend up their bilges to a perpendicular height of twice the depth of floor amidships from upper side of keel at middle line (vide b, Figs. 1 and 2), and not to be less moulded at their heads than the moulding of the frames. A floor plate to be fitted and rivetted to every frame; and to be worked across the middle line, except where centre through plates are adopted, so as to unite the sides of the vessel efficiently to each other. Watercourses are to be formed through all the floor plates on each side of middle line, so as to allow water to reach the pumps freely. (Vide Figs. 5 to 14.)

STRINGERS AND TIE PLATES.—All vessels to have stringer plates (of the thickness given in Table), upon the ends of each tier of beams. Those upon the ends of upper deck beams in vessels with two decks or tiers of beams, and on ends of middle deck beams in vessels with three decks or tiers of beams, to be in width one inch for every seven feet of the vessel's entire length, for half her length amidships, and from thence to the ends of the vessel they may be gradually reduced to three-fourths the width amidships—in no case, however, is the width to be less than eighteen inches amidships. The stringer plates are to be fitted home and rivetted to the outside plating at all upper decks, and at the middle deck in vessels having three decks, with angle iron of the dimensions given in Table (vide a, Fig. 1); the middle deck stringer plate to have an additional angle iron extending all fore and aft inside of the frames, rivetted to the reverse angle iron on the frames, and to the stringer plate (vide d, Figs. 1 and 2). Stringer plates on ends of beams below the upper deck in vessels with two decks, or below middle deck in vessels with three decks, may be reduced in width to three-fourths the midship breadth above named; this breadth is to be extended all fore and aft, and to have an angle iron of the dimensions given in Table, extended all fore and aft, rivetted to the reverse angle iron on the frames, and to the stringer plates (vide a, Figs. 1 and 2). In cases where no deck is laid, and the width of stringer plate on ends of hold beams is objected to, it may be reduced, provided such reduction be fully compensated for. The objectionable practice of cutting through the stringer plates for the admission of wood roughstair stanchions will not be allowed. All vessels are to have tie-plates ranging all fore and aft upon each tier of beams on each side of hatchways, and wherever practicable from side to side diagonally, to be in width once and a half the depth of beams, and of the thickness required for stringer plates (vide Fig. 15). The said plates are to be well rivetted to each other, and to beams, deck hooks, and transoms; and all butts to be properly shifted. Upon hold beams where no deck is to be laid, or where tie-plates would interfere with stowage of cargo, an angle iron of the dimensions given in Table for angle iron on beam stringers, placed at middle line, extending fore and aft where practicable, and well rivetted to all beams, deck hooks, and transoms, will be admitted in lieu thereof.

TABLE H.—(FROM LLOYD'S RULES.)

A TABLE exhibiting the different Descriptions of TIMBER, of Good Quality, to be used in the TIMBERING and PLANKING OF SHIPS, as the same will be applicable to the several Terms of Years appointed for Ships to remain on the Character A.

PARTS OF THE FRAME OF A VESSEL.		TWELVE YEARS.	NINE YEARS.	EIGHT YEARS.	SIX YEARS.	FIVE YEARS.	FOUR YEARS.
FLOORS.	English, African, and Live Oak, Adriatic, Italian, Spanish, Portuguese, and French Oak, East India Teak, Moring Saut, Greenheart, Morra, and Iron Bark.	The same as in the preceding Class, and admit—Mahogany of hard texture, Cuba Sabicu, Pencil Cedar, Angely, and Venetia.	The same as in the preceding Class, and admit—Other Continental White Oak, Spanish Chestnut, Blue Gum.	The same as in the preceding Class, and admit—North American White Oak, American Sweet Chestnut, Stringy Bark, Red Cedar.	The same as in the preceding Class, and admit—Pitch Pine, Larch, Hackmatack, Tamarac, Juniper, English Oak, African Oak, and § East India Teak.	The same as in the preceding Class, and admit—Cowlie, Huon Pine, American Rock Elm, Hickory.	The same as in the preceding Class, and admit—Baltic and American Red Pine, Foreign Ash, European and American Grey Elm, § B Birch, B. Walnut, Spruce Fir, White Chestnut, Beech.
1st FUTTOCKS.	English, African, and Live Oak, Adriatic, Italian, Spanish, Portuguese, and French Oak, East India Teak, Moring Saut, Greenheart, Morra, and Iron Bark.	The same as in the preceding Class, and admit—Mahogany of hard texture, Cuba Sabicu, Pencil Cedar, Angely, and Venetia.	The same as in the preceding Class, and admit—Other Continental White Oak, Spanish Chestnut, Blue Gum.	The same as in the preceding Class, and admit—North American White Oak, American Sweet Chestnut, Stringy Bark, Red Cedar.	The same as in the preceding Class, and admit—Pitch Pine, Larch, Hackmatack, Tamarac, Juniper, Secondhand English Oak, African Oak, and § East India Teak.	The same as in the preceding Class, and admit—Cowlie, Huon Pine, American Rock Elm, Hickory.	The same as in the preceding Class, and admit—Baltic and American Red Pine, Foreign Ash, European and American Grey Elm, § B Birch, B. Walnut, Spruce Fir, White Cedar, Beech.
2nd FUTTOCKS.	English, African, and Live Oak, Adriatic, Italian, Spanish, Portuguese, and French Oak, East India Teak, Moring Saut, Greenheart, Morra, and Iron Bark.	The same as in the preceding Class, and admit—Mahogany of hard texture, Cuba Sabicu, Pencil Cedar, Angely, and Venetia.	The same as in the preceding Class.	The same as in the preceding Class.	The same as in the preceding Class, and admit—Other Cont. White Oak, Spanish Chestnut, Blue Gum, N. Amer. White Oak, Amer. Sweet Chestnut, Stringy Bark, Red Cedar, Pitch Pine, Larch, Hackmatack, Tamarac, and Juniper.	The same as in the preceding Class, and admit—Secondhand English Oak, African Oak, § East India Teak, Cowlie, Huon Pine.	The same as in the preceding Class, and admit—Baltic and Amer. Red Pine, Foreign Ash, European and American Grey Elm, Black Birch and Black Walnut, Spruce Fir, White Cedar, and Hemlock.
3rd FUTTOCKS and TOP TIMBERS.	English, African, and Live Oak, Adriatic, Italian, Spanish, Portuguese, and French Oak, East India Teak, Moring Saut, Greenheart, Morra, and Iron Bark.	The same as in the preceding Class, and admit—Mahogany of hard texture, Cuba Sabicu, Pencil Cedar, Angely, and Venetia.	The same as in the preceding Class.	The same as in the preceding Class.	The same as in the preceding Class, and admit—Other Continental White Oak, Spanish Chestnut, Blue Gum, N. Amer. White Oak, Amer. Sweet Chestnut, Stringy Bark, Red Cedar, Pitch Pine, Larch, Hackmatack, Tamarac, Juniper, Cowlie, Huon Pine, Baltic and Amer. Red Pine.	The same as in the preceding Class, and admit—Secondhand English Oak, African Oak, § East India Teak, Cowlie, Huon Pine.	The same as in the preceding Class, and admit—Foreign Ash, European and American Grey Elm, Black Birch, Black Walnut, Spruce Fir, White Cedar, Yellow Pine, and Hemlock.
MAIN AND RIDER KEELSONS.	English, African, and Live Oak, Adriatic, Italian, Spanish, Portuguese, and French Oak, East India Teak, Moring Saut, Greenheart, Morra, and Iron Bark.	The same as in the preceding Class, and admit—Mahogany of hard texture, Cuba Sabicu, Pencil Cedar, Angely, and Venetia.	The same as in the preceding Class, and admit—Other Continental White Oak, Spanish Chestnut, Blue Gum.	The same as in the preceding Class, and admit—N. American White Oak, American Sweet Chestnut, Stringy Bark, Red Cedar, Pitch Pine, Larch, Hackmatack, Tamarac, Juniper.	The same as in the preceding Class, and admit—Other Cont. White Oak, Spanish Chestnut, Blue Gum, N. Amer. White Oak, Amer. Sweet Chestnut, Stringy Bark, Red Cedar, Pitch Pine, Larch, Hackmatack, Tamarac, and Juniper.	The same as in the preceding Class, and admit—English Ash, American Rock Elm, Hickory, Secondhand English Oak, African Oak, § East India Teak.	The same as in the preceding Class, and admit—Foreign Ash, European and American Grey Elm, Black Birch, Black Walnut, Spruce Fir, White Cedar, Beech, and Yellow Pine.
STEM AND STERN POSTS.	English, African, and Live Oak, Adriatic, Italian, Spanish, Portuguese, and French Oak, East India Teak, Moring Saut, Greenheart, Morra, and Iron Bark.	The same as in the preceding Class, and admit—Mahogany of hard texture, Cuba Sabicu, Pencil Cedar, Angely, and Venetia.	The same as in the preceding Class.	The same as in the preceding Class.	The same as in the preceding Class, and admit—Other Cont. White Oak, Spanish Chestnut, Blue Gum, N. Amer. White Oak, Amer. Sweet Chestnut, Stringy Bark, Red Cedar, Pitch Pine, Larch, Hackmatack, Tamarac, and Juniper.	The same as in the preceding Class, and admit—Cowlie, Huon Pine.	The same as in the preceding Class, and admit—Baltic and Amer. Red Pine, English Ash, Foreign Ash, European and American Grey Elm, Black Birch, Black Walnut, Spruce Fir, White Cedar, and Yellow Pine.
TRANSOMS, KNIGHT-HEADS, HAWSE-TIMBERS, APRON, and * DEADWOOD.	English, African, and Live Oak, Adriatic, Italian, Spanish, Portuguese, and French Oak, East India Teak, Moring Saut, Greenheart, Morra, and Iron Bark.	The same as in the preceding Class, and admit—Mahogany of hard texture, Cuba Sabicu, Pencil Cedar, Angely, and Venetia.	The same as in the preceding Class.	The same as in the preceding Class.	The same as in the preceding Class, and admit—Other Cont. White Oak, Spanish Chestnut, Blue Gum, N. Amer. White Oak, Amer. Sweet Chestnut, Stringy Bark, Red Cedar, Pitch Pine, Larch, Hackmatack, Tamarac, and Juniper.	The same as in the preceding Class, and admit—Cowlie, Huon Pine.	The same as in the preceding Class, and admit—Baltic and Amer. Red Pine, English Ash, Foreign Ash, European and American Grey Elm, Black Birch, Black Walnut, Spruce Fir, White Cedar, and Yellow Pine.
BEAMS and HOOKS.	English, African, and Live Oak, Adriatic, Italian, Spanish, Portuguese, and French Oak, East India Teak, Moring Saut, Greenheart, Morra, and Iron Bark.	The same as in the preceding Class.	The same as in the preceding Class.	The same as in the preceding Class, and admit—Other Cont. White Oak, Spanish Chestnut, Blue Gum, Larch, Hackmatack, Tamarac, and Juniper.	The same as in the preceding Class, and admit—North American White Oak, American Sweet Chestnut, Stringy Bark, Red Cedar, Cowlie, Huon Pine, and Baltic and American Red Pine.	The same as in the preceding Class, and admit—Secondhand English Oak, African Oak, § East India Teak.	The same as in the preceding Class, and admit—English Ash, Foreign Ash, American Rock Elm, Hickory, European and American Grey Elm.
KNEES.	English, African, and Live Oak, Adriatic, Italian, Spanish, Portuguese, and French Oak, East India Teak, Moring Saut, Greenheart, Morra, and Iron Bark.	The same as in the preceding Class.	The same as in the preceding Class.	The same as in the preceding Class, and admit—Other Cont. White Oak, Spanish Chestnut, Blue Gum, Pitch Pine, Larch, Hackmatack, Tamarac, and Juniper.	The same as in the preceding Class, and admit—North American White Oak, American Sweet Chestnut, Stringy Bark, Red Cedar, Cowlie, Huon Pine, Baltic and American Red Pine, Spruce Fir, and White Cedar.	The same as in the preceding Class, and admit—English Ash, Foreign Ash, American Rock Elm, Hickory, European and American Grey Elm.	The same as in the preceding Class, and admit—English Ash, Foreign Ash, European and American Grey Elm, Black Birch, Black Walnut, Spruce Fir, White Cedar, and Yellow Pine.
PARTS OF THE OUTSIDE OF A VESSEL.							
KEEL to the 1st FUTTOCK HEADS.	English, African, and Live Oak, Adriatic, Italian, Spanish, Portuguese, and French Oak, East India Teak, Moring Saut, Greenheart, Morra, and Iron Bark.	The same as in the preceding Class, and admit—Cowlie, Huon Pine, English Ash, Foreign Ash, American Rock Elm, Hickory, European and American Grey Elm, and § B Birch, Chow, Red Kralje, and Kapur.	The same as in the preceding Class, and admit—Baltic and American Red Pine.	The same as in the preceding Class.	The same as in the preceding Class.	The same as in the preceding Class, and admit—Spruce Fir, White Cedar, Yellow Pine.	The same as in the preceding Class, and admit—Hemlock.
1st FUTTOCK HEADS to LIGHT WATER-MARK.	English, African, and Live Oak, Adriatic, Italian, Spanish, Portuguese, and French Oak, East India Teak, Moring Saut, Greenheart, Morra, and Iron Bark.	The same as in the preceding Class, and admit—Mahogany of hard texture, Cuba Sabicu, Pencil Cedar, Angely, Venetia, other Continental White Oak, Spanish Chestnut, Blue Gum, N. Amer. White Oak, Amer. Sweet Chestnut, Stringy Bark, Red Cedar, Pitch Pine, Larch, Hackmatack, Tamarac, Juniper, Hickory, § Amer. Rock Elm, Europ. and Amer. Grey Elm, and Chow, Red Kralje, and Kapur.	The same as in the preceding Class, and admit—Cowlie, Huon Pine, Baltic and American Red Pine.	The same as in the preceding Class, and admit—American Rock Elm, European and American Grey Elm, Beech.	The same as in the preceding Class, and admit—English Ash, Foreign Ash, Black Birch, Black Walnut.	The same as in the preceding Class, and admit—Spruce Fir, White Cedar.	The same as in the preceding Class, and admit—Hemlock.
LIGHT WATER-MARK to WALES.	English, African, and Live Oak, Adriatic, Italian, Spanish, Portuguese, and French Oak, East India Teak, Moring Saut, Greenheart, Morra, and Iron Bark.	The same as in the preceding Class, and admit—Mahogany of hard texture, Cuba Sabicu, Pencil Cedar, Angely, and Venetia.	The same as in the preceding Class, and admit—Other Continental White Oak, Spanish Chestnut, Blue Gum.	The same as in the preceding Class, and admit—North American White Oak, American Sweet Chestnut, Stringy Bark, Red Cedar, Pitch Pine, Larch, Hackmatack, Tamarac, and Juniper.	The same as in the preceding Class, and admit—North American White Oak, American Sweet Chestnut, Stringy Bark, Red Cedar, Cowlie, Huon Pine, Baltic and American Red Pine.	The same as in the preceding Class, and admit—American Rock Elm, Hickory.	The same as in the preceding Class, and admit—European and American Grey Elm, Foreign Ash, Black Birch, Black Walnut, Beech, Hemlock.
WALES, BLACKSTRAKES, TOPSIDES, and SHEESTRAKES.	English, African, and Live Oak, Adriatic, Italian, Spanish, Portuguese, and French Oak, East India Teak, Moring Saut, Greenheart, Morra, and Iron Bark.	The same as in the preceding Class, and admit—Mahogany of hard texture, Cuba Sabicu, Pencil Cedar, Angely, and Venetia.	The same as in the preceding Class.	The same as in the preceding Class, and admit—Other Cont. White Oak, Spanish Chestnut, Blue Gum, Pitch Pine, Larch, Hackmatack, Tamarac, and Juniper.	The same as in the preceding Class, and admit—North American White Oak, American Sweet Chestnut, Stringy Bark, Red Cedar, Cowlie, Huon Pine, Baltic and American Red Pine.	The same as in the preceding Class.	The same as in the preceding Class, and admit—Europ. and Amer. Grey Elm, Black Birch, Black Walnut, Spruce Fir, White Cedar, and Hemlock.
UPPER DECK WATERWAYS, SPIRKETING, and FLANKSHEERS.	English, African, and Live Oak, Adriatic, Italian, Spanish, Portuguese, and French Oak, East India Teak, Moring Saut, Greenheart, Morra, and Iron Bark.	The same as in the preceding Class, and admit—Mahogany of hard texture, Cuba Sabicu, Pencil Cedar, Angely, Venetia, other Continental White Oak, Spanish Chestnut, Blue Gum, N. Amer. White Oak, Amer. Sweet Chestnut, Stringy Bark, Red Cedar, Pitch Pine, Larch, Hackmatack, Tamarac, Juniper, Cowlie, Huon Pine, and Baltic and American Red Pine.	The same as in the preceding Class, and admit—Other Continental White Oak, Spanish Chestnut, Blue Gum.	The same as in the preceding Class.	The same as in the preceding Class, and admit—North American White Oak, American Sweet Chestnut, Stringy Bark, Red Cedar.	The same as in the preceding Class.	The same as in the preceding Class, and admit—Europ. and American Grey Elm, Black Birch, Black Walnut, Spruce Fir, White Cedar, and Hemlock.
PARTS OF THE INSIDE OF A VESSEL.							
SHELVES, CLAMPS, LIMBER, BIGE STRAKES, GELLING IN HOLD, and DETWIST DECKS, also SPIRKETING and WATERWAY BELOW THE UPPER DECK.	English, African, and Live Oak, Adriatic, Italian, Spanish, Portuguese, and French Oak, East India Teak, Moring Saut, Greenheart, Morra, and Iron Bark.	The same as in the preceding Class, and admit—Mahogany of hard texture, Cuba Sabicu, Pencil Cedar, Angely, and Venetia.	The same as in the preceding Class, and admit—Other Continental White Oak, Spanish Chestnut, Blue Gum.	The same as in the preceding Class, and admit—North American White Oak, American Sweet Chestnut, Stringy Bark, Red Cedar.	The same as in the preceding Class, and admit—Pitch Pine, Larch, Hackmatack, Tamarac, Juniper, Cowlie, Huon Pine, Baltic and American Red Pine.	The same as in the preceding Class, and admit—American Rock Elm, Hickory.	The same as in the preceding Class, and admit—European and American Grey Elm, Foreign Ash, Black Birch, Black Walnut, Spruce Fir, White Cedar, and Hemlock.
GUDDER and WINDLASS MAIN PIECES.	English, African, and Live Oak, Adriatic, Italian, Spanish, Portuguese, and French Oak, East India Teak, Moring Saut, Greenheart, Morra, and Iron Bark.	The same as in the preceding Class, and admit—Mahogany of hard texture, Cuba Sabicu, Pencil Cedar, Angely, and Venetia.	The same as in the preceding Class.	The same as in the preceding Class.	The same as in the preceding Class, and admit—Other Continental White Oak, Spanish Chestnut, Blue Gum, N. Amer. White Oak, Amer. Sweet Chestnut, Stringy Bark, Red Cedar, Pitch Pine, Larch, Hackmatack, Tamarac, and Juniper.	The same as in the preceding Class.	The same as in the preceding Class, and admit—Baltic and American Rock Elm, Hickory, English Ash, Secondhand English Oak, African Oak, and § East India Teak.

OUTSIDE PLANKING

INSIDE PLANKING

* This Table applies to the Deadwood so far as regards the material brought to the surface.

1. This Table applies to the Deadwood as far as regards the material to be used from the height of two feet above the table top of the keel.

The use of Elm and Birch, in ships above the EIGHT YEARS' grade, to be restricted to a height from the lower part of the mainmast, of one-third of the internal depth of the ship.

The materials marked thus [under the head of "Materials"] are to be used in the construction of the internal depth of the Ship measured, in midships, from the top of the Upper Strake to the top of the Upper

* Black Birch, Birch, Amoria, a Black Elm, and Gum.

* Black Birch, Beech, American Rock Elm, and Cowditch
entire length of the keel in ships of the SEVEN YEARS' grade.

†† Yellow Pine allowed for Waterways of Upper Deck

^{††} White Cedar allowed for Third Futtocks and Top-timbers.

In cases where secondhand Peak of approved quality is proposed to be used, application may be made to the Committee with a view to its being allowed a higher grade (not exceeding two years) than as set forth above.

TABLES OF SCANTLINGS OF THE UNDERWRITERS' REGISTRY FOR IRON VESSELS, LIVERPOOL.

(EXTRACTED BY PERMISSION OF THE COMMITTEE.)

TABLE No 1.

TONNAGE.....	250	500	750	1000	1500	2000	2500	3000
Thickness of Centre Plate Keels.....	$\frac{1}{16}$	$\frac{9}{16}$	$\frac{1}{16}$	$\frac{1}{16}$	$\frac{1}{16}$	$\frac{1}{16}$	$\frac{1}{16}$	$\frac{1}{16}$
Size of Side Plates for Centre Plate Keels.....	$6 \times \frac{1}{16}$	$7 \times \frac{1}{16}$	$8 \times \frac{1}{16}$	$9 \times \frac{1}{16}$	$10 \times \frac{1}{16}$	$12 \times \frac{1}{16}$	$12 \times \frac{1}{16}$	$13 \times \frac{1}{16}$
Size of Bar Keels, in inches, Stems and Stern-posts.....	6×2	7×2	$8 \times 2\frac{1}{2}$	$9 \times 2\frac{1}{2}$	10×3	12×3	12×3	13×3
Lengths of Scarphs in Bar Keels, in inches.....	18	19	20	22	24	26	28	30
Diameter of Rudder Heads, in inches.....	$3\frac{1}{2}$	$4\frac{1}{2}$	5	6	$6\frac{1}{2}$	7	8	$8\frac{1}{2}$
Diameter of Rudder Heels and Pintles.....	2	$2\frac{1}{2}$	$2\frac{3}{4}$	3	$3\frac{1}{4}$	$3\frac{1}{2}$	4	$4\frac{1}{4}$
Diameter of Windlass Spindles.....	3	$3\frac{1}{2}$	$3\frac{3}{4}$	4	$4\frac{1}{2}$	5	$5\frac{1}{2}$	6

NOTE.—Rudder Heads in Screw Steamers, in all cases, to be one inch larger in diameter.

TABLE No. 2.

FRAMES.		Size of Reverse Frame and Beam Angle Irons, in Inches.	
Clear Depth of Hold.	Size of Frame, in Inches.	Size of Reverse Frame and Beam Angle Irons, in Inches.	Size of Reverse Frame and Beam Angle Irons, in Inches.
5 feet	$2\frac{1}{2} \times 2\frac{1}{2}$	2×2	$4-16ths$
10 feet	$3 \times 2\frac{1}{2}$	$2\frac{1}{2} \times 2$	$4-16ths$
14 feet	3×3	$2\frac{1}{2} \times 2\frac{1}{2}$	$5-16ths$
16 feet	$3\frac{1}{2} \times 3$	$2\frac{1}{2} \times 2\frac{1}{2}$	$6-16ths$
18 feet	4×3	$3 \times 2\frac{1}{2}$	$6-16ths$
20 feet	$4\frac{1}{2} \times 3$	$3\frac{1}{2} \times 3$	$6-16ths$
22 feet	5×3	$3\frac{1}{2} \times 3\frac{1}{2}$	$7-16ths$
24 feet	5×3	$3\frac{1}{2} \times 3\frac{1}{2}$	$7-16ths$
26 feet	$5\frac{1}{2} \times 3$	$4 \times 3\frac{1}{2}$	$8-16ths$
28 feet	$5\frac{1}{2} \times 3\frac{1}{2}$	$4 \times 3\frac{1}{2}$	$8-16ths$
30 feet	$6 \times 3\frac{1}{2}$	$4\frac{1}{2} \times 3\frac{1}{2}$	$9-16ths$

TABLE No. 3.

BEAMS AND FLOORS.		Centre Depth of Beam upon alternate Frames, in Inches.		Thickness, in Inches.
Breadth of Vessel.	Depth of Beam upon alternate Frames, in Inches.	Centre Depth of Beam upon alternate Frames, in Inches.	Thickness, in Inches.	Thickness, in Inches.
12 feet	$3\frac{1}{2}$	8		$3-16ths$
15 feet	4	10		$4-16ths$
20 feet	5	$13\frac{1}{2}$		$5-16ths$
22 feet	$5\frac{1}{2}$	$14\frac{1}{2}$		$6-16ths$
24 feet	6	16		$6-16ths$
26 feet	$6\frac{1}{2}$	$17\frac{1}{2}$		$7-16ths$
28 feet	7	$18\frac{1}{2}$		$7-16ths$
30 feet	$7\frac{1}{2}$	20		$8-16ths$
32 feet	8	21		$8-16ths$
34 feet	$8\frac{1}{2}$	$22\frac{1}{2}$		$9-16ths$
37 feet	$9\frac{1}{2}$	$24\frac{1}{2}$		$9-16ths$
40 feet	10	$26\frac{1}{2}$		$10-16ths$
44 feet	11	$28\frac{1}{2}$		$10-16ths$
48 feet	12	32		$11-16ths$

TABLE No. 4.

STANCHIONS FOR BEAMS.		TWIXT DECK STANCHIONS.	
Length of Ship.	Depth of Hold.	Length of Ship.	Depth of Hold.
6 feet hold		$2\frac{1}{2}$ inches diameter.	
7 " "		$2\frac{1}{2}$ " "	
8 " "		$2\frac{1}{2}$ " "	
9 feet hold		3 inches diameter.	
10 " "		$3\frac{1}{2}$ " "	
11 " "		$3\frac{1}{2}$ " "	
12 " "		$3\frac{1}{2}$ " "	
13 " "		$3\frac{1}{2}$ " "	
14 " "		$3\frac{1}{2}$ " "	
15 " "		$3\frac{1}{2}$ " "	
16 " "		$3\frac{1}{2}$ " "	

TABLE No. 5.

PLATES.		Thickness of Plates 8-5ths of Vessel Amidships, in Inches.		Thickness of Plates 1-5th from the ends of Vessel, in Inches.
Length of Ship.	Depth of Hold.	Thickness of Plates 8-5ths of Vessel Amidships, in Inches.	Thickness of Plates 1-5th from the ends of Vessel, in Inches.	Thickness of Plates 1-5th from the ends of Vessel, in Inches.
100 feet	10 feet	$6-16ths$		$5-16ths$
125 feet	$12\frac{1}{2}$ feet	$7-16ths$		$6-16ths$
150 feet	15 feet	$8-16ths$		$7-16ths$
175 feet	$17\frac{1}{2}$ feet	$9-16ths$		$8-16ths$
200 feet	20 feet	$10-16ths$		$9-16ths$
225 feet	22 feet	$10-16ths$		$9-16ths$
250 feet	24 feet	$10-16ths$		$9-16ths$
275 feet	26 feet	$11-16ths$		$10-16ths$
300 feet	28 feet	$11-16ths$		$10-16ths$

TABLE No. 6.

DECK TIES, KEELSONS, AND STRINGERS.		Length of Vessel, in Feet.		Breadth of Gunwale Stringer, in Inches.	Breadth of Lower Deck and Orip Stringer, in Inches.	Beam Tie-Plates each side of Hatches, all fore and aft, in Inches.	Depth of Centre Keelson above floors, in Inches.	Angle Irons on Keelsons, Deck Tie-Plates, and Stringers, in Inches.	Thickness, in Sixteenths of an Inch.
Length of Vessel, in Feet.	Breadth of Gunwale Stringer, in Inches.	Breadth of Lower Deck and Orip Stringer, in Inches.	Beam Tie-Plates each side of Hatches, all fore and aft, in Inches.	Depth of Centre Keelson above floors, in Inches.	Angle Irons on Keelsons, Deck Tie-Plates, and Stringers, in Inches.	Thickness, in Sixteenths of an Inch.			
100	25	18	$6\frac{1}{2}$	10	3×3	6			
120	26	20	$7\frac{1}{2}$	10	$3\frac{1}{2} \times 3$	7			
130	27	$20\frac{1}{2}$	9	12	$4 \times 3\frac{1}{2}$	7			
140	28	21	$9\frac{1}{2}$	12	$4 \times 3\frac{1}{2}$	7			
150	29	$21\frac{1}{2}$	10	14	4×4	8			
160	30	22	$10\frac{1}{2}$	14	4×4	8			
170	31	$22\frac{1}{2}$	11	16	4×4	8			
180	32	23	$11\frac{1}{2}$	16	$5 \times 3\frac{1}{2}$	9			
190	33	24	12	16	$5 \times 3\frac{1}{2}$	9			
200	34	25	$12\frac{1}{2}$	18	$5 \times 3\frac{1}{2}$	9			
210	35	26	13	18	$5 \times 3\frac{1}{2}$	9			
220	36	27	$13\frac{1}{2}$	19	5×4	10			
230	37	28	14	19	5×4	10			
240	38	29	$14\frac{1}{2}$	20	5×4	10			
250	39	30	15	20	5×4	10			
260	40	$30\frac{1}{2}$	$15\frac{1}{2}$	21	5×4	10			
270	41	31	16	21	5×4	11			
280	42	$31\frac{1}{2}$	16	22	5×4	11			
290	43	32	$16\frac{1}{2}$	22	5×4	11			
300	44	$32\frac{1}{2}$	17	23	5×4	11			
310	45	33	$17\frac{1}{2}$	23	5×4	11			
320	46	$33\frac{1}{2}$	18	24	5×5	12			
330	47	34	18	24	5×5	12			
340	48	$34\frac{1}{2}$	18	25	5×5	12			
350	49	36	18	25	6×5	12			
360	50	38	18	25	6×5	12			

TABLE No. 7.

DIAMETER OF RIVETS, IN SIXTEENTHS OF AN INCH.		THICKNESS OF PLATES, IN SIXTEENTHS OF AN INCH.	
8	10	5	6
12	13	7	8
13	14	8	9
14	15	9	10
15	16	10	11
16	18	11	12
18	19	12	13
19	20	13	14
20		14	15
		15	16

TABLE No. 8.

MASTS.		Length.		Diameter.	Body Thickness.	Head.	Angle Irons.
Length.	Diameter.	Body Thickness.	Head.	Angle Irons.			
60	20	$\frac{3}{8}$	$\frac{1}{8}$	$3 \times 2\frac{1}{2} \times \frac{3}{8}$			
72	24	$\frac{3}{8}$	$\frac{1}{8}$	$3 \times 3 \times \frac{3}{8}$			
78	26	$\frac{3}{8}$	$\frac{1}{8}$	$3 \times 4 \times \frac{3}{8}$			
84	28	$\frac{7}{8}$	$\frac{1}{8}$	$3 \times 4 \times \frac{1}{2}$			
90	30	$\frac{7}{8}$	$\frac{1}{8}$	$5 \times 3 \times \frac{1}{2}$			
96	32	$\frac{1}{2}$	$\frac{7}{8}$	$5 \times 3 \times \frac{1}{2}$			

YARDS OF STEEL.

YARDS OF STEEL.		Length.		Diameter.	Centre.	To Arms.	Arms.	Angle Irons.
Length.	Diameter.	Centre.	To Arms.	Arms.	Angle Irons.			
60	15	$\frac{1}{2}$	$\frac{1}{8}$	$\frac{3}{8}$				
64	16	$\frac{1}{2}$	$\frac{1}{8}$	$\frac{3}{8}$				
68	17	$\frac{1}{2}$	$\frac{3}{8}$	$\frac{1}{4}$				
72	18	$\frac{1}{2}$	$\frac{1}{8}$	$\frac{3}{8}$				
76	19	$\frac{1}{2}$	$\frac{1}{8}$	$\frac{3}{8}$				
80	20	$\frac{1}{2}$	$\frac{1}{8}$	$\frac{3}{8}$				
84	21	$\frac{1}{2}$	$\frac{1}{8}$	$\frac{3}{8}$				

All angles in masts and yards to extend the whole length.

TABLE No. 10.

REQUIRED ATTESTED PROOF ON CHAINS.

Same as Table No. 22, see page 170.

BOATS.

Vessels above 400 tons to have at least three boats, viz:—long boat, pinnace and gig. Vessels under 400 tons to have at least two boats, viz:—long boat and pinnace.

COMPASSES.

Each Vessel to be supplied with not less than—One Steering Compass (Compensated) with spare card, and spare pivots and centres—One Standard Azimuth Compass, with spare card, and spare pivots and centres, placed so as to command the horizon over the weather side when the ship is heeling.

NOTE.—To insure good action in the Compasses of iron vessels each Card should have two parallel needles, strongly magnetized, fixed so that their similar poles may be 60° apart. The point of suspension must be in the plane of the gimball. Pivots to be of steel, hardened and tempered to a straw colour, or to have iridium points. Centres to be of ruby or chrysolite. A convenient size for the card of a steering compass is 9 inches in diameter, and for an azimuth compass 7 inches in diameter.

DIVISION FOURTH.

PRACTICAL SHIPBUILDING.

CHAPTER I.

SPECIAL PROPERTIES OF MATERIALS.

SECTION I.—IRON AND STEEL.

ARTICLE 1. *Sources and Kinds of Iron.*—The following are the most common conditions in which iron is found in its ores:—

	By Chemical equivalents.*	By Weight.	Per centage of Iron.
I. <i>Native Iron</i> , being iron nearly pure, or combined with from one-fourth to one-hundredth part of its weight of nickel. This is found in detached masses, and is very rare,.....			80 to 100
II. Protoxide or Black Oxide of Iron,.....	{Iron,..... 2 ... 56}	{ 72 ... 77·8	
	{Oxygen,... 1 ... 16}		
Protoxide of iron is found only in combination with other substances.			
III. Peroxide or Red Oxide of Iron,.....	{Iron,..... 4 ... 112}	{ 160 ... 70	
	{Oxygen,... 3 ... 48}		
IV. Magnetic Oxide of Iron,...	{Iron,..... 3 ... 84}	{ 116 ... 72·4	
	{Oxygen,... 2 ... 32}		
V. Hydrate of Peroxide of Iron =			
Peroxide of Iron, 2 equiv.,	{Iron,..... 8 ... 224}	{ 374 ... 60	
	{Oxygen,... 6 ... 48}		
Water,.....3 equiv.,	{Oxygen,... 3 ... 144}		
	{Hydrogen, 6 ... 6}		
VI. Carbonate of Iron =			
Protoxide of Iron, 1 equiv.,	{Iron,..... 2 ... 56}	{ 116 ... 48·3	
	{Oxygen,... 1 ... 16}		
Carbonic Acid,....1 equiv.,	{Oxygen,... 2 ... 48}		
	{Carbon,... 2 ... 12}		

Iron is found combined with sulphur, forming what is called *Iron Pyrites*; but that mineral is not available for the manufacture of iron; and it forms a pernicious ingredient in ores, or in the fuel used to smelt them, because of the weakening effect of sulphur upon iron. The same is the case with *Phosphate of Iron*.

The most abundant foreign ingredients found mixed with compounds of iron in its ores are siliceous sand, and silicate of alumina, or clay; next in abundance are the carbonates of lime and magnesia. Amongst other foreign ingredients, which, though not abundant, have an influence on the quality of the iron produced, are carbon, manganese, arsenic, &c. Of these manganese and carbon alone are beneficial: all the rest are hurtful.

The most common *Ores of Iron* are the following:—

I. **MAGNETIC IRON ORE**, consisting of magnetic oxide of iron,

pure, or almost pure, and containing 72 per cent. of iron, is found chiefly in veins traversing the primary strata, and amongst plutonic rocks, and is the source of some of the finest qualities of iron, such as those of Sweden and the North-eastern United States.

II. **RED IRON ORE** is peroxide of iron, pure or mixed. When pure and crystalline, it is called *Specular Iron Ore*, or *Iron-glance*; when pure, or nearly so, and in kidney-shaped masses, showing a fibrous structure, it is called *Red Hæmatite*; when mixed with less or more clay and sand, it is called *Red Ironstone* and *Red Ochre*. It is found in various geological formations, and is purest in the oldest. The purer kinds, iron-glance and hæmatite, produce excellent iron; for example, that of Nova Scotia.

III. **BROWN IRON ORE** is hydrate of peroxide of iron, pure or mixed. When compact and nearly pure, it is called *Brown Hæmatite*; when earthy and mixed with much clay, *Yellow Ochre*. It is found amongst various strata, especially those of later formations.

IV. **CARBONATE OF IRON**, when pure and crystalline, is called *Sparry* or *Spathose Iron Ore*; when mixed with clay and sand, *Clay Ironstone*; when clay ironstone is coloured black by carbonaceous matter, it is called *Black-band Ironstone*. These ores are found amongst various primary and secondary stratified rocks, and especially amongst those of the coal formation. The proportion of earthy matter in the ordinary ores containing carbonate of iron ranges from 10 to 40 per cent.

The iron of Britain is manufactured partly from hæmatite, but chiefly from clay ironstone and black-band.

The metallic products of the iron manufacture are of three kinds; *malleable* or *wrought iron*, being pure or nearly pure iron; *cast iron* and *steel*, being certain compounds of iron with carbon.

2. *Impurities of Iron.*—The strength and other good qualities of those products depend mainly on the absence of impurities, and especially of certain substances which are known to cause brittleness and weakness, of which the most important are, sulphur, phosphorus, silicon, calcium, and magnesium.

Sulphur and calcium, and probably also magnesium, make iron "*red short*," that is, brittle at high temperatures; phosphorus and silicon make it "*cold short*," that is, brittle at low temperatures. These are both serious defects; but the latter is the worse.

* The chemical equivalents adopted in the above Table, are as follows:—

Oxygen,	16
Carbon,	6
Hydrogen,	1
Iron,	28

Sulphur comes in general from coal or coke used as fuel. Its pernicious effects can be avoided altogether by using fuel which contains no sulphur; and hence the strongest and toughest of all iron is that which is smelted, reduced, and puddled either with charcoal, or with coal or coke that is free from sulphur.

Phosphorus comes in most cases from phosphate of iron in the ore, or from phosphate of lime in the ore, the fuel, or the flux. The ores which contain most phosphorus are those found in strata where animal remains abound, such as those of the oolitic formation.

Calcium and *Silicon* are derived respectively from the decomposition of lime and of silica by the chemical affinity of carbon for their oxygen. The only iron which is entirely free from these impurities is that which is made by the reduction of ores that contain neither silica, nor lime, such as pure magnetic iron ore, pure hæmatite, and pure sparry iron ore.

If either of those earths be present in the ore, the other must be added as a *flux*, to form a slag with it; and a small portion of each of them will be deoxidated, the bases uniting with the iron.

The statements made relative to calcium, are applicable also to magnesium.

The effect of aluminium upon iron is not known with certainty.

3. *Cast Iron* is the product of the process of *smelting* iron ores. In that process the ore in fragments, mixed with fuel and with flux, is subjected to an intense heat in a blast-furnace, and the products are *slag*, or glassy matter formed by the combination of the flux with the earthy ingredients of the ore, and *pig iron*, which is a compound of iron and carbon, either unmixed, or mixed with a small quantity of uncombined carbon in the state of plumbago.

The ore is often *roasted* or calcined before being smelted, in order to expel carbonic acid and water.

The proportions of ore, fuel, and flux are fixed by trial; and the success of the operation of smelting depends much on those proportions. The flux is generally limestone, from which the carbonic acid is expelled by the heat of the furnace; while the lime combines with the silica and alumina of the ore. If the ore contains carbonate of lime, less lime is required as a flux. If either lime or silica is present in excess, part of the earth which is in excess forms a glassy compound with oxide of iron, which runs off amongst the slag, so that part of the iron is wasted; and another part of that earth becomes reduced, its base combining with the iron and making it brittle, as has been stated in the preceding Article; so that in order to produce at once the greatest quantity and best quality of iron from the ore, the earthy ingredients of the entire charge of the furnace must be in certain definite proportions, which are discovered for each kind of ore by careful experiment.

The total quantity of carbon in pig iron ranges from 2 to 5 per cent. of its weight.

Different kinds of pig iron are produced from the same ore in the same furnace under different circumstances as to temperature and quantity of fuel. A high temperature and a large quantity of fuel produce *grey cast iron*, which is further distinguished into No. 1, No. 2, No. 3, and so on; No. 1 being that produced at the highest temperature. A low temperature and a deficiency of fuel produce *white cast iron*. Grey cast iron is of different shades of bluish-grey in colour, granular in texture, softer and more easily fusible than white cast iron. White cast iron is silvery white, either granular or crystalline, comparatively difficult to melt, brittle, and excessively hard.

It appears that the differences between those kinds of iron

depend, not so much on the total quantities of carbon which they contain, as on the proportions of that carbon which are respectively in the conditions of mechanical mixture and of chemical combination with the iron. Thus, grey cast iron contains *one* per cent., and sometimes less, of carbon in chemical combination with the iron, and from *one to three* or *four* per cent. of carbon in the state of plumbago in mechanical mixture; while white cast iron is a homogeneous chemical compound of iron with from 2 to 4 per cent. of carbon. Of the different kinds of grey cast iron, No. 1 contains the greatest proportion of plumbago, No. 2 the next, and so on.

There are two kinds of white cast iron, the *granular* and the *crystalline*. The granular kind can be converted into grey cast iron by fusion and slow cooling; and grey cast iron can be converted into granular white cast iron by fusion and sudden cooling. This takes place most readily in the best iron. Crystalline white cast iron is harder and more brittle than granular, and is not capable of conversion into grey cast iron by fusion and slow cooling. It is said to contain more carbon than granular white cast iron; but the exact difference in their chemical composition is not yet known.

Grey cast iron, No. 1, is the most easily fusible, and produces the finest and most accurate castings; but it is deficient in hardness and strength; and, therefore, although it is the best for castings of moderate size, in which accuracy is of more importance than strength, it is inferior to the harder and stronger kinds, No. 2 and No. 3, for large structures.

The presence of plumbago renders iron comparatively weak and pliable, so that the order of strength and stiffness among different kinds of cast iron from the same ore and fuel is as follows:—

Granular white cast iron.
Grey cast iron, No. 3.
“ “ No. 2.
“ “ No. 1.

Crystalline white cast iron is not introduced into this classification, because its extreme brittleness makes it unfit for use in engineering structures or in machinery.

Granular white cast iron also, although stronger and harder than grey cast iron, is too brittle to be a safe material for the entire mass of any girder, or other large piece of a structure or machine; but it is used to form a hard and impenetrable *skin* to a piece of grey cast iron by the process called *chilling*. This consists in lining the portion of the mould where a hardened surface is required with suitably shaped pieces of iron. The melted metal, on being run in, is cooled and solidified suddenly where it touches the cold iron; and for a certain depth from the chilled surface, varying from about $\frac{1}{8}$ th to $\frac{1}{2}$ inch in different kinds of iron, it takes the white granular condition, while the remainder of the casting takes the grey condition.

Even in castings which are not chilled by an iron lining to the mould, the outermost layer, being cooled more rapidly than the interior, approaches more nearly to the white condition, and forms a *skin* harder and stronger than the rest of the casting.

The best kinds of cast iron for large structures are No. 2 and No. 3; because, being stronger than No. 1, and softer and more flexible than white cast iron, they combine strength and pliability in the manner which is best suited for safely bearing loads that are in motion.

A strong kind of cast iron called *toughened cast iron*, is produced by the process invented by Mr. Morries Stirling, of adding to the

cast iron, and melting amongst it, from one-fourth to one-seventh of its weight of wrought iron scrap.

Malleable Cast Iron is made by the following process:—The castings to be made malleable are imbedded in the powder of red hæmatite; they are then raised to a bright red heat (which occupies about 24 hours), maintained at that heat for a period varying from three to five days, according to the size of the casting, and allowed to cool (which occupies about 24 hours more). The oxygen of the hæmatite extracts part of the carbon from the cast iron, which is thus converted into a sort of soft steel; and its tenacity (according to experiments by Messrs. A. More & Son) becomes more than 48,000 lbs. per square inch.

The strength of cast iron to resist cross breaking was found by Mr. Fairbairn to be increased by *repeated meltings* up to the *twelfth*, when it was greater than at first in the ratio of 7 to 5 nearly. After the twelfth melting that sort of strength rapidly fell off.

The resistance to crushing went on increasing after each successive melting; and after the *eighteenth* melting it was double of its original amount, the iron becoming silvery white and intensely hard.

The transverse strength of No. 3 cast iron was found by Mr. Fairbairn not to be diminished by raising its temperature to 600° Fahr. (being about the temperature of melting lead). At a red heat its strength fell to two-thirds.

The strength of cast iron of every kind is marked by two properties; the smallness of the tenacity as compared with the resistance to crushing, and the different values of the modulus of rupture of the same kind of iron in bars torn directly asunder, and in beams of different forms when broken across.

For the results of experiments on the strength of various kinds of cast iron, see the Table at the end of Division III.

4. *Castings for Engineering Structures and Machinery*.—The best course for an engineer or shipbuilder to take, in order to obtain cast iron of a certain strength, is not to specify to the founder any particular kind or mixture of pig iron, but to specify a certain minimum strength which the iron should show when tested by experiment.

As to the appearance of good iron for engineering castings, it should show on the outer surface a smooth, clear, and continuous skin with regular faces and sharp angles. When broken, the surface of fracture should be of a light bluish-grey colour and close-grained texture, with considerable metallic lustre; both colour and texture should be uniform, except that near the skin the colour may be somewhat lighter and the grain closer; if the fractured surface is *mottled*, either with patches of darker or lighter iron, or with crystalline spots, the casting will be unsafe; and it will be still more unsafe if it contains air-bubbles. The iron should be soft enough to be slightly indented by a blow of a hammer on an edge of the casting.

Castings are tested for air-bubbles by ringing them with a hammer all over the surface.

Cast iron, like many other substances, when at or near the temperature of fusion, is a little more bulky for the same weight in the solid than in the liquid state, as is shown by the solid iron floating on the melted iron. This causes the iron as it solidifies to fill all parts of the mould completely, and to take a sharp and accurate figure:

The solid iron contracts in cooling from the melting point down

to the temperature of the atmosphere, by $\frac{1}{8}$ th part in each of its linear dimensions, or *one-eighth of an inch in a foot*; and therefore patterns for castings are made larger in that proportion than the intended pieces of cast iron which they represent.

In designing patterns for castings, care must be taken to avoid all abrupt variations in the thickness of metal, lest parts of the casting near each other should be caused to cool and contract with unequal rapidity, and so to split asunder or overstrain the iron.

Iron becomes more compact and sound by being cast under pressure; and hence cast iron cylinders, pipes, columns, and the like, are stronger when cast in a vertical than in a horizontal position, and stronger still when provided with a *head*, or additional column of iron, whose weight serves to compress the mass of iron in the mould below it. The air bubbles ascend and collect in the head, which is broken off when the casting is cool.

Care should be taken not to cut or remove the skin of a piece of cast iron at those points where the stress is intense.

Cast iron expands in linear dimensions by about $\frac{1}{800}$, or '0011, in rising from the freezing to the boiling point of water; being at the rate of '00000617 for each degree of Fahrenheit's scale, or about '0004 for the range of temperature which is usual in the British climate. Every structure containing cast iron must be so designed that the greatest expansion and contraction of the castings by change of temperature shall not injure the structure.

5. *Wrought or Malleable Iron* in its perfect condition is simply pure iron. It falls short of that perfect condition to a greater or less extent owing to the presence of impurities, of which the most common and injurious have been mentioned, and their effects stated, in Article 2; and its strength is in general greater or less according to the greater or less purity of the ore and fuel employed in its manufacture.

Malleable iron may be made either by direct reduction of the ore, or by the abstraction of the carbon and various impurities from cast iron. The process of direct reduction is applicable to rich and pure ores only; and it leaves a slag or "cinder" which contains a large proportion of oxide of iron, and yields pig iron by smelting. The most economical and generally applicable process is that of removing the foreign constituents from pig iron; and for that purpose white pig iron (called "forge pig") is usually employed, partly because it retains less carbon on the whole than grey pig iron, and partly because it is unfit for making castings. The details of the process are very much varied; but the most important principle of its operation always is to bring the pig iron in a melted state into close contact with a quantity of air sufficient to oxidate all the carbon and silicon. The carbon escapes in carbonic oxide or carbonic acid gas; the silica produced by the oxidation of the silicon combines partly with protoxide of iron and partly with lime (which is sometimes introduced as a flux for it), and forms slag or "cinder." Chloride of sodium (common salt) is used to remove sulphur and phosphorus. In another form of the process this is performed by injecting jets of steam amongst the molten iron; the oxygen of the steam assists in oxidating the carbon and silicon, and the hydrogen combines with the sulphur and phosphorus. The surest method, however, of obtaining iron free from the weakening effects of sulphur and phosphorus is to employ ores and fuel that do not contain those constituents.

The most common form of the process of making malleable iron is *puddling*, in which the pig iron is melted in a reverberatory furnace, and is brought into close contact with the air by stirring

it with a rake or "rabble." Some iron-makers precede the process of puddling by that of "refining," in which the pig iron, in a melted state, has a blast of air blown over its surface. This removes part of the carbon, and leaves a white crystalline compound of iron and carbon called "refiners' metal." Others omit the refining, and at once puddle the pig iron; this is called "*pig boiling*." The removal of the carbon is indicated by the thickening of the mass of iron, malleable iron requiring a higher temperature for its fusion than cast iron. It is formed into a lump called a "loup" or "bloom," taken out of the furnace, and placed under a tilt hammer or in a suitable squeezing machine, to be "*shingled*;" that is, to have the cinder forced out, and the particles of iron welded together by blows or pressure.

The bloom is then passed between rollers, and rolled into a bar; the bar is cut into short lengths, which are fagotted together, reheated, and rolled again into one bar; and this process is repeated till the iron has become sufficiently compact, and has acquired a fibrous structure.

Bars are called No. 1, No. 2, No. 3 bars, &c., according to the number of times they have been rolled.

In Mr. Bessemer's process, the molten pig iron, having been run into a suitable vessel, has jets of air blown through it by a blowing machine. The oxygen of the air combines with the silicon and carbon of the pig iron, and in so doing produces enough of heat to keep the iron in a melted state till it is brought to the malleable condition; it is then run into large ingots, which are hammered and rolled in the usual way. The process has been most successful when applied to pig iron that is free from sulphur and phosphorus, such as that of Sweden and Nova Scotia.

Strength and toughness in bar iron are indicated by a fine, close, and uniform fibrous structure, free from all appearance of crystallization, with a clear bluish-grey colour and silky lustre on a torn surface where the fibres are shown.

Plate iron of the best kind consists of alternate layers of fibres crossing each other. It should have a hard smooth skin, somewhat glossy, and when broken, should show perfect uniformity of structure, and be free from all tendency to split into layers.

To examine the internal structure of iron, whether in bars or in plates, a short piece may be notched on one side near the middle and bent double. The fitness of bar iron for shipbuilding and smith-work is tested by bending and punching it cold, and by punching and forging it hot, so as to ascertain whether it shows any signs of brittleness either when cold or when hot, (called "cold-short" and "hot-short").

Malleable iron is distinguished by the property of *welding*: two pieces, if raised nearly to a white heat and pressed or hammered firmly together, adhering so as to form one piece. In all operations of which welding forms a part, such as rolling and forging, it is essential that the surfaces to be welded should be brought into close contact, and should be perfectly clean and free from oxide of iron, cinder, and all foreign matter.

In all cases in which several bars are to be fagotted or rolled into one, attention should be paid to the manner in which they are "*piled*" or built together, so that the pressure exerted by the hammer or the rollers may be transmitted through the whole mass. If this be neglected, the finished bar, plate, or other piece, may show flaws marking the divisions between the bars of the pile.

Wrought iron, although it is at first made more compact and strong by *reheating* and hammering, or otherwise working it, soon

reaches a state of maximum strength, after which all reheating and working rapidly makes it weaker, as is shown in the Tables of Division III. Good bar iron has in general attained its maximum strength; and therefore, in all operations of forging it, whether on a great or small scale, by the steam-hammer or by that in the hand of the blacksmith, the desired size and figure ought to be given with the least possible amount of reheating and working.

It is of great importance to the strength of all pieces of forged iron that the *continuity of the fibres* near the surface should be as little interrupted as possible; in other words, that the fibres near the surface should lie in layers parallel to the surface.*

Another important principle in designing pieces of forged iron which are to sustain shocks and vibrations, is to avoid as much as possible abrupt variations of dimensions, and angular figures, especially those with re-entering angles; for at the points where such abrupt variations and angles occur fractures are apt to commence. If two parts of a shaft, for example, or of a beam exposed to shocks and vibrations, are to be of different thicknesses, they should be connected by means of curved surfaces, so that the change of thickness may take place gradually, and without re-entering angles.

6. *Steel and Steely Iron*.—Steel, the hardest of the metals and the strongest of known substances, is a compound of iron with from 0·5 to 1·5 per cent. of its weight of carbon. These, according to most authorities, are the only essential constituents of steel.

The term "steely iron," or "semi-steel," may be applied to compounds of iron with less than 0·5 per cent. of carbon. They are intermediate in hardness and other properties between steel and malleable iron.

In general, such compounds are the harder and the stronger, and also the more easily fusible, the more carbon they contain; those kinds which contain less carbon, though weaker, are more easily welded and forged, and from their greater pliability are the fitter for structures that are exposed to shocks.

Impurities of different kinds affect steel injuriously in the same way with iron.

There are certain foreign substances which have a beneficial effect on steel. One 2000th part of its weight of silicon causes steel to cool and solidify without bubbling or agitation; but a larger proportion is not to be used, as it would make the steel brittle. The presence of manganese in the iron, or its introduction into the crucible or vessel in which steel is made, improves the steel by increasing its toughness and making it easier to weld and forge.

Steel is distinguished by the property of *tempering*; that is to say, it can be hardened by sudden cooling from a high temperature, and softened by gradual cooling; and its degree of hardness or softness can be regulated with precision by suitably fixing that temperature. The ordinary practice is, to bring all articles of steel to a high degree of hardness by sudden cooling, and then to soften them more or less by raising them to a temperature which is the higher the softer the articles are to be made, and letting them cool very gradually. The elevation of temperature previous to the "annealing" or gradual cooling is produced by plunging the articles into a bath of a fusible metallic alloy. The temperature of the bath ranges from 430° to 560° Fahr.

* On this subject see a paper by the Editor of this Treatise, in the Proceedings of the Institution of Civil Engineers for 1846.

According to the experiments of Mr. Kirkaldy, a great increase of strength is produced by hardening steel in oil.

Steel is made by various processes, which have of late become very numerous. They may all be classed under two heads, viz., adding carbon to malleable iron, and abstracting carbon from cast iron. The former class of processes, though the more complex, laborious, and expensive, is preferred for making steel for cutting tools and other fine purposes, because of its being easier to obtain malleable iron than cast iron in a high state of purity. The latter class of processes is the best adapted for making great masses of steel and steely iron rapidly and at moderate expense. The following are some of the processes employed in making different kinds of steel:—

Blister Steel is made by a process called "*cementation*," which consists in imbedding bars of the purest wrought iron (such as that manufactured by charcoal from magnetic iron ore) in a layer of charcoal, and subjecting them for several days to a high temperature. Each bar absorbs carbon, and its surface becomes converted into steel, while the interior is in a condition intermediate between steel and iron. Cementation may also be performed by exposing the surface of the iron to a current of carburetted hydrogen gas at a high temperature. Cementation is sometimes applied to the surfaces of articles of malleable iron in order to give them a skin or coating of steel, and is called "*case-hardening*."

Shear Steel is made by breaking bars of blister steel into lengths, making them into bundles or fagots, and rolling them out at a welding heat, and repeating the process until a near approach to uniformity of composition and texture has been obtained. It is used for various tools and cutting implements.

Cast Steel is made by melting bars of blister steel in a crucible, along with a small additional quantity of carbon (usually in the form of coal tar) and some manganese. It is the purest, most uniform, and strongest steel, and is used for the finest cutting implements.

Another process for making cast steel, but one requiring a higher temperature than the preceding, is to melt bars of the purest malleable iron with manganese and with the whole quantity of carbon required in order to form steel. The quality of the steel as to hardness is regulated by the proportion of carbon. A sort of semi-steel, or steely iron, made by this process, and containing a small proportion of carbon only, is known as *homogeneous metal*.

Steel made by the air blast is produced from molten pig iron by Mr. Bessemer's process in two ways; either the blowing of jets of air through the iron is stopped at an instant determined by experience, when it is known that a quantity of carbon still remains in the iron sufficient to make steel of the kind required, or else the blast is continued until the carbon is all removed, so that the vessel is full of pure malleable iron in the melted state, and carbon is added in the proper proportion, along with manganese and silicon. The usual way of adding the carbon is by running into the vessel a sufficient quantity of highly carbonized cast iron. The steel thus produced is run into large ingots, which are hammered and rolled like blooms of wrought iron.

Puddled Steel is made by puddling pig iron, and stopping the process at the instant when the proper quantity of carbon remains. The bloom is shingled and rolled like bar iron.

Granulated Steel is made by running melted pig iron into a cistern of water, over a wheel, which dashes it about so that it is

found at the bottom of the cistern in the form of grains or lumps of the size of a hazel nut, or thereabouts. These are imbedded in pulverized hæmatite, or in sparry iron ore, and exposed to a heat sufficient to cause part of the oxygen of the ore to combine with and extract the carbon from the superficial layer of each of the lumps of iron, each of which is reduced to the condition of malleable iron at the surface, while its heart continues in the state of cast iron. A small additional quantity of malleable iron is produced by the reduction of the ore. These ingredients, being melted together, produce steel.

There are other processes for making steel and steely iron, of which the details are not yet publicly known.

7. Strength of Wrought Iron and Steel.—Wrought iron, like fibrous substances in general, is more tenacious along than across the fibres; and its tenacity is greater than its resistance to crushing. The effect of the latter difference on the best forms of cross-section for beams has already been considered in the Third Division.

The ductility of wrought iron often causes it to yield by degrees to a load, so that it is difficult to determine its strength with precision.

Wrought iron has its longitudinal tenacity considerably increased by rolling and wire-drawing; so that the smaller sizes of bars are on the whole more tenacious than the larger; and iron wire is more tenacious still, as is shown in the Table of tenacity at the end of Division III.

Wrought iron is weakened by too frequent reheating and forging; so that even in the best of large forgings, the tenacity is only about *three-fourths* of that of the bars from which the forgings were made, and sometimes even less.

As to the *effect of heat on the strength of wrought iron*, it has been shown by Mr. Fairbairn ("*Useful Information for Engineers*," second series)—

I. That the tenacity of ordinary *boiler plate* is not appreciably diminished at a temperature of 395° Fahr., but that at a dull red heat it is diminished to about three-fourths.

II. That the tenacity of good *rivet iron* increases with elevation of temperature up to about 320° Fahr., at which point it is about one-third greater than at ordinary atmospheric temperatures; and that it then diminishes, and at a red heat is reduced to little more than one-half of its value at ordinary atmospheric temperatures.

Numerous experiments have been made on the tenacity of steel; but its other kinds of strength have been very little investigated. Its tenacity, like that of bar iron, is increased by rolling and wire-drawing.

Plate iron is somewhat less tenacious crosswise than lengthwise; but the difference ought not to exceed about one-tenth.

When the tenacity of iron intended for purposes of shipbuilding is tested by a machine, it is considered not to be fit for use if the specimen is broken by a less load than 20 tons on the square inch of the original sectional area, or 24 tons on the square inch of the area as diminished by drawing out at the place of fracture. As to the greater strength possessed by the better qualities of iron, see the Tables.

It is highly important also that the iron should possess *toughness*; and this may be tested by observing *in what proportion the length of the piece is increased at the instant before breaking*. The ultimate elongation of really good and tough specimens of iron and

steel, as ascertained in Mr. Kirkaldy's experiments, was nearly as follows, in fractions of the original length :—

Bar iron, from.....	0.15	to 0.30
Plate iron, lengthwise, from.....	0.04	to 0.17
" crosswise, from.....	0.015	to 0.11
Steel bars, from.....	0.05	to 0.19
Steel plates, from.....	0.03	to 0.19

8. *Preservation of Iron in the Air.*—The present Article has reference only to the preservation of pieces of iron work exposed to the air. The subject of the preservation of ships' bottoms belongs to a later Chapter of the present Division; and that of the preservation of steam-boilers, to the Sixth Division.

The corrosion of iron is a sort of slow combustion, during which the iron combines with oxygen, and produces rust. The ordinary methods of preserving iron in the air consist principally in preventing the access of oxygen to the metal.

Cast iron will often last for a long time without rusting, if care be taken not to injure its skin, which is usually coated with a film of silicate of the protoxide of iron, produced by the action of the sand of the mould on the iron. Chilled surfaces of castings are without that protection, and therefore rust more rapidly.

The corrosion of iron is more rapid when partly wet and partly dry, than when wholly immersed in water or wholly exposed to the air. It is accelerated by impurities in water, and especially by the presence of decomposing organic matter, or of free acids. It is also accelerated by the contact of iron with any metal which is electro-negative relatively to the iron, or in other words, has less affinity for oxygen (such as copper), or with the rust of the iron itself. If two portions of a mass of iron are in different conditions, so that one has less affinity for oxygen than the other, the contact of the former makes the latter oxidate more rapidly. In general, hard and crystalline iron is less rapidly oxidable than ductile and fibrous iron. Cast iron and steel decompose rapidly in warm or impure sea-water.

Pieces of iron which are kept constantly in a state of vibration oxidate less rapidly than those which are at rest.

(See Mallet "On the Corrosion of Iron," in the "Reports of the British Association" for 1843 and 1849).

The following are amongst the ordinary methods of preserving iron which is not immersed in sea-water, hot or cold, nor exposed to hot steam :—

I. Boiling in coal-tar, especially if the pieces of iron have first been heated to the temperature of melting lead.

II. Heating the pieces of iron to the temperature of melting lead, and smearing their surfaces, while hot, with cold linseed oil, which dries and forms a sort of varnish.

III. Painting with oil-paint, which must be renewed from time to time. The linseed oil process is a good preparation for painting.

IV. Coating with zinc, commonly called galvanizing. This is efficient, provided it is not exposed to acids capable of dissolving the zinc; but it is destroyed by sulphuric acid in the atmosphere of places where much coal is burned.

9. *Expansion of Iron and Steel by Heat.*—The following are proportionate longitudinal expansions produced by heating from the temperature of melting ice (32° Fahr., or 0° Cent.) to that of water boiling under the average atmospheric pressure (212° Fahr., or 100° Cent.), being a rise of 180° Fahr. = 100° Cent.

Cast iron.....	0.00111
Steel.....	0.00120
Malleable iron.....	0.00125

SECTION II.—COPPER AND OTHER METALS AND ALLOYS.

10. *Copper* (except when found in the native or metallic state) is extracted from ores which contain that metal in the state of oxide, of carbonate, or of sulphuret; and the sulphuret of copper is almost always in combination with sulphuret of iron, and often with sulphurets and selenurets of arsenic, antimony, lead, and other metals. Besides the copper-bearing minerals, most of the ores contain earthy matter, consisting chiefly of silica and alumina.

The richer ores containing oxides and carbonates are in the first place smelted along with a flux (usually lime) suitable for combining with the earthy constituents, which run off in the form of slag; the greater part of the oxygen is taken away by the carbon of the fuel; the substance remaining, called *black copper*, is an impure copper, containing small quantities of iron, oxide of iron, silica, sulphur, and other ingredients. The ores containing sulphurets are roasted, sometimes repeatedly, before being melted, to expel part of the sulphur; the principal product obtained after smelting is called *matt*, and is a compound of sulphuret of copper with sulphuret of iron.

The poorer ores containing oxides and carbonates are sometimes deoxidated by being smelted along with sulphuret of iron; the iron takes the oxygen, and the copper the sulphur, and the products are matt and black copper mixed.

The matt obtained by either of those processes is subjected to a series of smeltings along with a siliceous flux, which have the effect of gradually expelling the sulphur, and removing the iron in the form of a silicate of the protoxide of iron; so that the final product is black copper.

The black copper is purified by melting it in a suitable furnace, in which a stream of air is directed on its surface; the oxygen of the air combines with and carries off the iron, sulphur, and other impurities. The melted copper, when purified as completely as possible, retains a certain quantity of oxygen, which is removed as far as possible by sprinkling the surface of the melted metal with charcoal, and stirring it with a pole of green wood. Much skill is required to stop this process at the right moment; for whether it is stopped too soon, or carried on too long, the metal obtained is injured and weakened; in the former case by the presence of oxide of copper; and in the latter, by some weakening action of the carbon, the precise nature of which is not known with certainty.

The greater or less admixture of oxide of copper with the metal, is indicated by the greater or less redness of its colour.

The preceding is a very brief outline of the processes of smelting and refining copper ores; for the details of which, as well as for an account of various recent modifications and improvements, reference must be made to works on metallurgy, such as those of Mr. Phillips and Dr. Percy.

For marine purposes, copper is chiefly used in the form of sheets for sheathing ships, and bolts for fastening them, made by rolling and hammering; which operations considerably increase its tenacity (see the Table of the Strength of Metals).

The quality principally required in such copper is that of resisting the oxidating action of sea-water. At the same time, it must not be wholly protected against oxidation; but must be eaten away at the slowest rate that is sufficient to make a thin film of oxide scale off from time to time, carrying with it the shell-fish and weeds which would otherwise incrust the bottom of the ship and impede her motion.

The protection of ships' bottoms will be treated of fully in a later Chapter; the present Article is limited to the consideration of the qualities of copper suitable for sheathing. Our knowledge of that subject is still very imperfect. A summary of the facts hitherto ascertained regarding it is given in a paper read by Mr. W. J. Hay, Admiralty Chemist, to the Institution of Naval Architects in 1863 (see their Transactions for that year). From that paper it appears that the following conclusions may be regarded as established:—

I. Extreme purity of the copper, as regards freedom from other metals, is not essential to durability: on the contrary, the following proportions of alloy are favourable to durability:—

Silver, from.....	$\frac{1}{100}$ to $\frac{1}{1000}$.
Iron, ".....	$\frac{1}{100}$ " $\frac{1}{100}$.
Zinc, ".....	$\frac{1}{10}$ " $\frac{1}{100}$.
Tin, ".....	$\frac{1}{100}$ " $\frac{1}{100}$.

Potassium, in the proportion of $\frac{1}{100}$, softens copper.

II. Carbon, in the proportion of from $\frac{1}{100}$ to $\frac{1}{1000}$, combined chemically with copper, renders it hard. A larger proportion of carbon does not combine chemically with the copper, but mixes with it mechanically, and is very injurious.

III. Oxygen and sulphur in the copper are very injurious.

IV. *Uniformity* of texture and composition is of the highest importance to durability; and nothing tends more to promote corrosion of the copper than the presence of patches of electro-negative metal or of oxide; because those patches, together with the other parts of the copper and the sea-water, form electric circuits.

As a means of increasing the compactness, and consequently the durability, of sheet copper for sheathing, *cold rolling* is recommended by Mr. Fincham.

11. *Alloys of Copper with Tin and with Zinc* are used, like copper itself, for sheathing and fastenings, and also for machinery. Various names, such as *brass*, *bronze*, *mixed metal*, &c., are applied to them indiscriminately; but strictly speaking, *Bronze* is the proper name of the alloys of copper and tin; *Brass*, that of the alloys of copper and zinc.

Both classes of alloys are less expensive than pure copper; and that is one reason for using them. Bronze, besides, is at least equal to copper in tenacity, and is considerably superior in hardness and resistance to crushing. Brass is inferior to copper in strength. Both bronze and brass make good castings, which quality is not possessed by copper.

As zinc is cheaper than tin, alloys of copper and zinc are preferable to those of copper and tin for sheathing: their inferiority of strength being of little consequence for that purpose.

The following general principle should be observed in the manufacture of all alloys whatsoever, as being essential to the soundness, strength, and durability of the compound metal—*The quantities of the constituents should bear definite atomic proportions to each other.*

For example, the chemical equivalents of Copper, Tin, and Zinc bear to each other the following proportions:—

Copper.	Tin.	Zinc.
31.5	59	32.5
or 63	118	65

and the proportions in which they are combined in any alloy should be expressed by multiples of those numbers.

When this rule is not observed, the metal produced is not a homogeneous compound, but a mixture of two or more different compounds in irregular masses, shown by a mottled appearance

when broken; and those masses being different in expansibility and elasticity, tend to separate from each other; and being different in chemical composition, they produce electric circuits and promote corrosion.

The following is a list of some of the alloys of copper with tin and zinc, which are fit for use in machinery or in shipbuilding.

COMPOSITION.				ALLOYS OF COPPER AND TIN.
By Equivalents.		By Weight.		
Copper.	Tin.	Copper.	Tin.	
12	1	378	59	Very hard bronze.
14	1	441	59	Hard bronze, for machinery bearings.
16	1	504	59	{ Bronze, or gun-metal: contracts in cool-
18	1	567	59	ing from its melting point, $\frac{1}{10}$.
20	1	630	59	Bronze, somewhat softer.
				Soft bronze, for toothed wheels, &c.

COMPOSITION.				ALLOYS OF COPPER AND ZINC.
By Equivalents.		By Weight.		
Copper.	Zinc.	Copper.	Zinc.	
4	1	126	32.5	Malleable brass.
2	1	63	32.5	{ Ordinary brass: contracts in cooling from its melting point, $\frac{1}{10}$.
3	2	94.5	65	{ Yellow metal for sheathing and fasten- ings. This is cast in ingots, and rolled and worked at a red heat into sheets and bolts.

Various alloys of copper, tin, and zinc, are used in machinery, and may be regarded as modifications of true bronze, produced by substituting one or two equivalents of zinc for one or two equivalents of the copper. They are less expensive than true bronze, but not so tough.

12. *Other Alloys.*—The strongest of all alloys yet known is *Aluminium Bronze*, as a reference to the Table of the strength of metals will show. Different sorts contain from 5 to 10 per cent. of aluminium, and from 95 to 90 per cent. of copper; and if 31.5 be taken as the equivalent of copper, and 13.7 as that of aluminium, their atomic constitution is probably from 8 to 4 equivalents of copper to one equivalent of aluminium.

Alloys of copper and lead, called *pot-metal*, are sometimes used for cocks and valves where strength is unimportant; but they are weak and brittle; and in bronze, lead is a pernicious adulteration.

Soft metal, for the bearings of shafts, consists of 50 parts of tin, 1 of copper, and 5 of antimony; it is, in fact, a sort of metallic grease. Its use will be referred to in the Sixth Division.

SECTION III.—TIMBER.

13. *Structure of Timber.*—Timber is the material of trees belonging almost exclusively to that class of the vegetable kingdom in which the stem grows by the formation of successive layers of wood all over its external surface, and is therefore said by botanists to be *exogenous*.

The exceptions are, trees of the palm family, and tree-like grasses, such as the bamboo, which belong to the *endogenous* class; so called because, although the stem grows partly by the formation of layers of new wood on its outer surface, the fibres of that new wood do nevertheless cross and penetrate amongst those previously formed in such a manner as to be mixed with them in one part of their course, and internal to them at another.

The stems of endogenous trees, though light and tough, are too flexible and slender to furnish materials suitable for shipbuilding. They will therefore not be further mentioned in this Section, except to refer to the Tables in the Third Division for the tenacity and heaviness of bamboo.

The stem of an exogenous tree is covered with bark, which grows by the formation of successive layers on its inner surface, at the same time that the wood grows by the formation of successive layers on its outer surface. This double operation takes place in the narrow space between the previously-formed wood and bark, during the circulation of the sap. The sap ascends from the roots to the leaves through vessels contained in the outer layers of the wood; at the surface of the leaves it acquires carbon from the atmosphere, and becomes denser, thicker, and more complex in its composition; it then descends from the leaves to the roots through vessels contained chiefly in the innermost layers of the bark. It is believed that the formation of new wood and bark takes place either wholly or principally from the descending sap.

The circulation of the sap is either wholly or partially suspended during a portion of each year (in tropical climates during the dry season, and in temperate and polar climates during the winter); and hence the wood and bark are usually formed in distinct layers, at the rate of one layer in each year; but this rule is not universal. Each such layer consists of parts differing in density and colour to an extent which varies in different kinds of trees.

The tissues of which both wood and bark consist are distinguished into two kinds—*cellular tissue*, consisting of clusters of minute cells; and *vascular tissue*, or *woody fibre*, consisting of bundles of slender tubes; the latter being distinguished from the former by its fibrous appearance. The difference, however, between those two kinds of tissue, although very distinct both to the eye and to the touch, is really one of degree rather than of kind; for the fibres or tubes of vascular tissue are simply very much elongated cells, tapering to points at the ends, and “breaking joint” with each other.

The tenacity of wood when strained “along the grain” depends on the tenacity of the walls of those tubes or fibres; the tenacity of wood when strained “across the grain” depends on the adhesion of the sides of the tubes and cells to each other. Examples of the difference of strength in those different directions are given in the Tables.

When a woody stem is cut across, the cellular and vascular tissue are seen to be arranged in the following manner:—

In the centre of the stem is the *pith*, composed of cellular tissue, inclosed in the medullary sheath, which consists of vascular tissue of a particular kind. From the pith there extend, radiating outwards to the bark, thin partitions of cellular tissue, called *medullary rays*; between these, additional medullary rays extend inwards from the bark to a greater or less distance, but without penetrating to the pith.

When the medullary rays are large and distinct, as in oak, they are called “*silver grain*.”

Between the medullary rays lie bundles of vascular tissue, forming the woody fibre, arranged in nearly concentric rings or layers round the pith. These rings are traversed radially by the medullary rays. The boundary between two successive rings is marked more or less distinctly by a greater degree of porosity, and by a difference of hardness and colour.

The annual rings are usually thicker at that side of the tree which has had most air and sunshine, so that the pith is not exactly in the centre.

The wood of the entire stem may be distinguished into two parts—the outer and younger portion, called “*sap-wood*,” being softer, weaker, and less compact, and sometimes lighter in colour,

than the inner and older portion, called “*heart-wood*.” The heart-wood is alone to be employed in those works of carpentry in which strength and durability are required. The boundary between the sap-wood and the heart-wood is in general distinctly marked, as if the change from the former to the latter occurred in the course of a single year. The following examples of the proportion of sap-wood to the entire volume are given on the authority of Tredgold (“Principles of Carpentry,” Section X.):—

Tree.	Age. Years.	Diameter. Inches.	Rings of Sap-wood.	Thickness of Sap-wood. Inches.	Proportion of Sap-wood to whole Trunk.
Chestnut.....	58	16½	7	⅘	0.1
Oak.....	65	17	17	1½	0.294
Scotch Fir.....	?	24	?	2½	0.416

The following data are given on the authority of Mr. Andrew Murray, C.E. (Ency. Brit., article “Timber”):—

Tree.	Rings of Sap-wood.
English Oak (<i>Quercus pedunculata</i>).....	12 to 15
Durmast Oak (<i>Quercus sessiliflora</i>).....	20 to 30
Chestnut (<i>Castanea Vesca</i>).....	5 or 6
Elm (<i>Ulmus campestris</i>).....	about 10
Larch (<i>Larix Europæa</i>).....	“ 15
Scotch Fir (<i>Pinus sylvestris</i>).....	“ 30
Memel Fir (<i>Pinus sylvestris</i>).....	“ 44
Canadian Yellow Pine (<i>Pinus variabilis</i>).....	“ 42

The structure of a *branch* is similar to that of the trunk from which it springs, except as regards the difference in the number of annual rings, corresponding to the difference of age. A branch becomes partially imbedded in those layers of the trunk which are formed after the time of its first sprouting; it causes a perforation in those layers, accompanied by distortion of their fibres, and constitutes what is called a *knot*. (On various matters mentioned in this Article, see Balfour’s “Manual of Botany,” Part I., chaps. i. and ii.)

14. *Timber Trees Classified*.—For purposes of carpentry, trees may be classed according to the mechanical structure of the wood. It has already been stated that the botanical classes of Endogens and Exogens correspond to essential differences of mechanical structure.

In further dividing the class of exogenous trees, or timber-trees proper, according to the structure of the wood, a division into two classes at once suggests itself, which exactly corresponds with a botanical division, viz.:—

Pine-wood, comprising all timber-trees belonging to the coniferous order; and

Leaf-wood, comprising all other timber-trees.

Beyond this primary division, the place of a tree in the botanical system has little or no connection with the structure of its timber.

A classification of timber according to its mechanical structure was proposed by Tredgold, founded, in the first place, on the greater or less distinctness of the medullary rays; and secondly, on the greater or less distinctness of the annual rings. According to that classification, pine-wood, or coniferous timber, is placed in the same class with leaf-wood that has the medullary rays indistinct; and this is certainly a fault in the system. If, however, pine-wood be placed in a class apart, Tredgold’s system may very well be applied to divide and subdivide the class of leaf-wood; but it is to be observed that the characters on which that system is founded, being mere differences in degree, and not in kind, are not of that definite sort which a thoroughly satisfactory system of classification requires; and if they are adopted, it is because no better set of distinguishing characters has yet been proposed.

The following is a condensed view of the classification of exogenous timber, as above described:—

CLASS I.—PINE-WOOD. (Natural order *Conifera*.)—Examples: Pine, Fir, Larch, Cowrie, Yew, Cedar, Juniper, Cypress, &c.

CLASS II.—LEAF-WOOD. (Non-coniferous trees.)

DIVISION I.—With distinct large medullary rays. (The trees in this division form part of the natural order *Amentacea*.)

Subdivision 1.—Annual rings distinct.—Example: Oak.

Subdivision 2.—Annual rings indistinct.—Examples: Beech, Plane, Sycamore, &c.

DIVISION II.—No distinct large medullary rays.

Subdivision 1.—Annual rings distinct.—Examples: Chestnut, Ash, Elm, &c.

Subdivision 2.—Annual rings indistinct.—Examples: Mahogany, Walnut, Teak, Greenheart, Mora, Lignum-Vita, &c.

The chief practical bearings of the foregoing classification are as follows:—

Pine-wood, or coniferous timber, in most cases, contains turpentine. It is distinguished by straightness in the fibre and regularity in the figure of the trees; qualities favourable to its use for spars, beams, and planking. The lightness of pine-wood makes the stronger kinds specially suitable for spars. At the same time, the lateral adhesion of the fibres is small; so that it is much more easily shorn and split along the grain, or torn asunder across the grain, than leaf-wood; and is therefore less fitted to resist thrust or shearing stress, or any kind of stress that does not act along the fibres. Even the toughest kinds of pine-wood are easily wrought. A peculiar characteristic of pine-wood (but one which requires the microscope to make it visible) is that of having the vascular tissue "*punctated*;" that is to say, there are small lenticular hollows in the sides of the tubular fibres. This structure is probably connected with the smallness of the lateral adhesion of those fibres to each other. Pine-wood is, on the whole, inferior to leaf-wood for the frames and skins of ships; because the strong kinds (as pine and fir) are deficient in durability; and the durable kinds (as cedar and cypress) are deficient in strength.

In *Leaf-wood*, or non-coniferous timber, there is no turpentine. The degree of distinctness with which the structure is seen, whether as regards medullary rays or annual rings, depends on the degree of difference of texture of different parts of the wood. Such difference tends to produce unequal shrinking in drying; and consequently those kinds of timber in which the medullary rays, and the annual rings, are distinctly marked, are more liable to warp than those in which the texture is more uniform. At the same time, the former kinds of timber are, on the whole, the more flexible, and in many cases are very tough and strong, which qualities make them suitable for structures that have to bear shocks.

15. *Appearance of good Timber*.—There are certain appearances which are characteristic of strong and durable timber, to what class soever it belongs.

In the same species of timber, that specimen will in general be the strongest and the most durable which has grown the slowest, as shown by the narrowness of the annual rings.

The cellular tissue as seen in the medullary rays (when visible) should be hard and compact.

The vascular or fibrous tissue should adhere firmly together, and should show no woolliness at a freshly-cut surface, nor should it clog the teeth of the saw with loose fibres.

If the wood is coloured, darkness of colour is in general a sign of strength and durability.

The freshly-cut surface of the wood should be firm and shining,

and should have somewhat of a translucent appearance. A dull, chalky appearance is a sign of bad timber.

In wood of a given species, the heavier specimens are in general the stronger and the more lasting.

Amongst resinous woods, those which have least resin in their pores, and, amongst non-resinous woods, those which have least sap or gum in them, are in general the strongest and most lasting.

Timber should be free from such blemishes as "clefts," or cracks radiating from the centre; "cup-shakes," or cracks which partially separate one annual layer from another; "upsets," where the fibres have been crippled by compression; "rind-galls," or wounds in a layer of the wood, which have been covered and concealed by the growth of subsequent layers over them; and hollows or spongy places, in the centre or elsewhere, indicating the commencement of decay.

16. *Examples of Pine-wood*.—The following are examples of timber of this class:—

I. *PINE* timber of the best sort is the produce of the Red Pine, or Scottish Fir (*Pinus sylvestris*), grown in Norway, Sweden, Russia, and Poland. The best is exported from Riga, the next from Memel and from Dantzic. The same species of tree grows also in Britain, but is inferior in strength. The annual rings, when this timber is of the best kind, consist of a hard part, of a clear dark-red colour, and a less hard part, of a lighter colour, but still clear and compact. The thickness of the rings should not exceed one-tenth of an inch. The most common size of the logs to be met with in the market, is about 13 inches square. This is the best of all timber for the spars of ships.

Pine timber is also obtained from various other species, chiefly North American, of which the best are the Yellow Pine (*Pinus variabilis*), and White Pine (*Pinus Strobus*). It is softer and less durable than the Red Pine of the North of Europe, but lighter, and can be had in larger logs.

II. *WHITE FIR*, or *DEAL* timber of the best kind, is the produce of the Spruce Fir (*Abies excelsa*), grown in Norway, Sweden, and Russia. The best is that known as Christiania Deal. Much of this timber is sawn up for sale into pieces of various thicknesses suited for planking, which,

When 7 inches broad are called	"battens."
When 9 " " "	"deals."
When 11 " " "	"planks."

They are to be had of various lengths; but the most usual length is about 12 feet.

This is an excellent kind of timber for boarding, light framing, and joiners' work, and for the lighter spars of ships.

Amongst other kinds of spruce fir, applied to the same purposes, are the North American White Spruce (*Abies alba*), and Black Spruce (*Abies nigra*).

III. The *LARCH* (*Larix Europæa*), grown in various parts of Europe, furnishes timber of great strength, and remarkable for durability when exposed to the weather; but harder to work and more subject to warp than red pine. The best sort has the harder part of the rings of a dark-red, and the softer part of a honey-yellow; and its rings are somewhat thicker than those of red pine.

Two North American species, the Black Larch, or Hackmatack (*Larix pendula*), and the Red Larch (*Larix microcarpa*), produce timber similar to that of the European Larch.

IV. The COWRIE or KAWRIE (*Dammara Australis*), a coniferous tree, grown in New Zealand, produces timber similar in its properties to the best kinds of pine, except that it is said to be more liable to warp, and more variable in quality. It is of a brownish-yellow colour, and more uniform in its texture than red pine and larch.

V. The term CEDAR is applied, not only to the timber of the true Cedar (*Cedrus Libani*), but also to that of various large species of Juniper (such as *Juniperus Virginiana*) and of Cypress. Those kinds of wood are remarkable for durability, in which they excel all other timbers; but they are deficient in strength.

17. *Examples of Leaf-wood with large Medullary Rays.*—The kinds of timber mentioned in this Article belong to the first division of Tredgold's system. Of the examples cited, the Oak alone belongs to the first subdivision, in which the divisions between the annual rings are distinctly marked by circles of pores. The other examples belong to the second subdivision, in which the rings are less distinctly marked:—

I. OAK timber, the strongest, toughest, and most lasting of those grown in temperate climates, is the produce of various species or varieties of the botanical genus *Quercus*. In Europe there are two kinds of oak trees; and it is doubtful whether they are distinct species or varieties of one species. They are—

The old English Oak, or Stalk-fruited Oak (*Quercus Robur* or *Quercus pedunculata*), in which the acorns grow on stalks, and the leaves close to the twig, and

The Bay Oak, or Cluster-fruited Oak (*Quercus sessiliflora*), in which the acorns grow in close clusters, and the leaves have short stalks.

Both those kinds of oak come to their greatest perfection in Britain.

The wood of the stalk-fruited oak is lighter in colour, and has more numerous and distinct medullary rays than that of the cluster-fruited oak, in which they are sometimes so few and indistinct as to have caused it in old buildings to be mistaken for chestnut. The stalk-fruited oak is the stiffer and the straighter-grained of the two, the easier to work, and the less liable to warp; it is therefore preferable where stiffness and accuracy of form are desired; the cluster-fruited oak is the more flexible, which gives it an advantage where shocks have to be borne.

The best oak timber when new is of a pale brownish-yellow colour, with a perceptible shade of green, a firm and glossy surface, very small and regular annual rings, and hard and compact medullary rays. Thick rings, many large pores, a dull surface, and a reddish, or "foxy" hue (caused by a fungus called "drux"), are signs of weak and perishable wood.

It is considered that oak timber comes to maturity at the age of 100 years, at which period each tree produces on an average about 75 cubic feet of timber; and that it should not be felled before the 60th year of its age, nor later than the 200th.

The species of oak in North America are very numerous. The best of them are, the Red Oak (*Quercus rubra*), and White Oak (*Quercus alba*), which are little inferior to the best European kinds, and the Live Oak (*Quercus virens*), which is said to be superior in strength, toughness, and durability, to all other species, but is very scarce. Large quantities of oak timber, very hard and durable, and well suited on account of great curvature for the frame-timbers of ships, have been obtained for H.M.

service from Italy; but as the timber on becoming seasoned is full of shakes, it is unsuitable for other purposes.

The wood of the oak contains gallic acid, which contributes to the durability of the timber, but corrodes iron fastenings. Metal fastenings for oak should therefore be of copper, or its alloys.

The following are examples of trees belonging to the second subdivision:—

II. BEECH (*Fagus sylvatica*), common in Europe.

III. AMERICAN PLANE (*Platanus occidentalis*), common in North America.

IV. SYCAMORE (*Acer Pseudo-platanus*), also called Great Maple, and in Scotland and the North of England, Plane; common in western Europe.

All these afford compact timber of uniform texture. They are valuable for blocks of wood which have to resist a crushing force, as for wedges for purposes of shipbuilding. They last well when constantly wet (especially beech, which is considered suitable for keels and bottom planking), but when alternately wet and dry they decay rapidly.

18. *Leaf-wood without large medullary rays.*—The examples of timber in this Article belong to the first subdivision of the second division, according to Tredgold's system, having no large distinct medullary rays, and having the divisions between the annual rings distinctly marked by a more porous structure. They are in general strong, but flexible.

I. The ASH (*Fraxinus excelsior*) furnishes timber whose toughness and flexibility render it superior to that of all other European trees for making handles of tools, shafts of carriages, and the like; but which is not sufficiently stiff and durable to be used in great works of carpentry. The colour of the wood is like that of oak, but darker, and with more of a greenish hue; the annual rings are broader than those of oak, and the difference between their compact and porous parts more marked.

II. The common ELM (*Ulmus campestris*) and Smooth-leaved Elm (*Ulmus glabra*) yield timber which is valued for its durability when constantly wet, and is specially suited for keels, garboard strakes, and bottom planking; but not for planking that is alternately wet and dry. Its strength across the grain, and its resistance to crushing, are comparatively great; and these properties render it useful for some parts of mechanism, such as shells of ships' blocks. There are other European species of elm, such as the Wych Elm (*Ulmus montana*), but their timber is inferior to that of the two species named.

A North American species, the Rock Elm, is said to be not only durable under water, but straight-grained and tough, so as to be well suited for long beams and stringers.

19. *Examples of Leaf-wood without large medullary rays continued.*—The kinds of timber mentioned in this Article are examples of the second subdivision of Tredgold's second division, having no large distinct medullary rays, and no distinct difference of compactness in the rings. This uniformity of structure is accompanied by comparative freedom from warping.

I. MAHOGANY (*Swietenia Mahagoni*) is produced in Central America and the West Indian Islands: that of the former region being commonly known as "Bay Mahogany;" that of the latter as "Spanish Mahogany." When of good quality, it is very straight-grained, very strong in all directions (though easily split along the grain), very durable, and preserves its shape under varying circumstances as to heat and moisture better than any

other kind of timber which can be procured in equal abundance. Mahogany varies much in quality; bay mahogany being in general superior to Spanish mahogany in strength, stiffness, and durability, and in the size of the logs, which are from 24 to 48 inches square. Bay mahogany of good quality is an excellent timber for all purposes of shipbuilding. Spanish mahogany is the more highly valued for ornamental purposes.

Spanish mahogany is distinguished by having a white chalky substance in its pores, those of bay mahogany being empty.

II. TEAK (*Tectona grandis*), from its great strength, stiffness, toughness, and durability, is the most valuable of all woods for ship-building. It is produced in the mountainous districts of south-eastern Asia and the East India Islands. The best comes from Moulmein, Malabar, Ceylon, Johore, and Java. The best logs of teak range from 40 to 80 feet in length, and from 15 to 30 inches square; trees are to be found of much greater size, but they are liable to be unsound at the heart. This timber is frequently perforated with holes to a considerable depth by insects, which defect frequently causes great loss in converting logs into planks.

Good teak resembles oak in colour and lustre, is very uniform and compact in texture, and has very narrow and regular annual rings. It contains a resinous, oily matter in its pores, in order to extract which the tree is sometimes tapped; but this injures the strength and durability of the timber, and ought to be avoided. Iron is not corroded by contact with teak, unless it has been grown in a marshy soil.

III. GREENHEART (*Nectandra Rodiei*), a tree of British Guiana, yields a very strong and durable timber, considered of the first quality for shipbuilding and all kinds of carpentry, and also for piled foundations and other structures under water. The colour is olive green, verging on drab and on black in two varieties respectively. The black variety is considered the more durable, but is very scarce. The lighter-coloured variety is comparatively abundant, and may be had in logs from 12 to 24 inches square, and from 40 to 70 feet long. Greenheart is very straight-grained, and its fibrous structure very distinct. The texture of the wood closely resembles that of bay mahogany.

IV. MORA (*Mora excelsa*), also a tree of British Guiana, yields a first-class timber for shipbuilding. The trunk grows very long and straight, and furnishes logs about as large as those of greenheart. The branches grow crooked, and are serviceable for knees and curved timbers. It is strong in all directions, and difficult to split or splinter.

V. LIGNUM-VITÆ (*Guaiacum officinale*) is produced in the West India Islands. It is remarkable for heaviness, compactness, toughness, and hardness, and for the property of resisting a crushing force with nearly equal strength across and along the grain; a property which makes it specially useful for rollers, sheaves, and dead-eyes. In converting logs into sheaves, the direction of the fibre of the timber is parallel to the axis of the sheave. The heart-wood is yellowish-green, the sap-wood greenish-yellow; and it is considered advisable, in cutting it into pieces suitable for sheaves and dead-eyes, to leave a ring of sap-wood all round the heart-wood, which is thus protected against too rapid drying, and prevented from splitting.

VI. AFRICAN TEAK, or AFRICAN OAK, called also TURTOSA, grows in the western tropical parts of Africa. It somewhat resembles true Teak in appearance, but has more of a brownish or

yellowish hue. It is strong, hard, and straight-grained, and is well suited for long and straight, or nearly straight, pieces, such as keelsons, stringers, shelf-pieces, waterways, and beams. It is often much damaged by shakes, and by the boring of insects.

VII. SABICU, a West Indian timber, is much esteemed for all parts of shipbuilding, being in general strong, durable, and sound. The best logs are straight-grained; others, through having grown spirally, have the fibres crooked and short, and are thereby unfitted to bear any great strain, except direct compression. Logs are sometimes found to be shaken and fractured at the heart, while the outside is sound: this is ascribed by Fincham to shocks received in felling or transportation.

20. *Leaf-wood continued—Australian Timber.*—Many species of the genus *Eucalyptus*, peculiar to Australia, yield timber of great size, strength, and durability; especially that of the IRON-BARK, BLUE-GUM, and JARRAH. The wood of iron-bark is white or yellowish; that of blue-gum, straw-coloured; that of jarrah resembles mahogany, and is sometimes called "Australian Mahogany." The *Eucalypti*, in common with some other Australian trees, are distinguished from the trees of other quarters of the globe by being more easily split in concentric layers, than in planes radiating from the pith; and the most frequent blemish in their timber is the occurrence of cracks of that kind, or "cup-shakes," filled with gum.

21. *Influence of Soil and Climate on Timber.*—Most timber trees are capable of flourishing in a great variety of soils. The best soil for all of them is one which, without being too dry and porous, allows water to escape freely, such as gravel mixed with sandy loam.

The most injurious soil to trees is that of swampy ground containing stagnant water: it never fails to make the timber weak and perishable.

As to the influence of climate, two general laws seem to prevail: that the strongest timber is yielded, amongst *different species* of trees, by those produced in tropical climates; and amongst trees of *the same species*, by those grown in cold climates. The first law is exemplified in such woods as teak, iron-wood, ebony, and lignum-vitæ, surpassing in strength all those of temperate climates: the second, in the red pine of Norway, as compared with that of Scotland, in the oak of Britain as compared with that of Italy, and even in the oak of Scotland and the North of England, as compared with that of the South of England.

22. *Age and Season for felling Timber.*—There is a certain age of maturity at which each tree attains its greatest strength and durability. If cut down before that age, the tree, besides being smaller, contains a greater proportion of sap-wood, and even the heart-wood is less strong and lasting; if allowed to grow much beyond that age, the centre of the tree begins either to become brittle, or to soften, and a decay commences by slow degrees, which finally renders the heart hollow. The age of maturity is therefore the best age for felling the tree to produce timber. The following data respecting it are given on the authority of Tredgold:—

	Age of Maturity. Years.
Oak,.....	60 to 200
Ash, Elm, Larch,.....	average 100
Fir.....	50 to 100
	70 to 100

The best season for felling timber is that during which the sap is not circulating—that is to say, in cold and temperate

climates, the winter, and in tropical climates, the dry season; for the sap tends to decompose, and so to cause decay of the timber. The best authorities recommend, also, as a means of hardening the sap-wood, that the bark of trees which are to be felled should be stripped off in the preceding spring.

Immediately after timber has been felled, it should be *squared*, by sawing off four "slabs" from the log, in order to give the air access to the wood and hasten its drying. If the log is large enough, it may be sawn into halves or quarters.

23. *Seasoning, Natural and Artificial.*—Seasoning timber consists in expelling, as far as possible, the moisture which is contained in its pores.

Natural Seasoning is performed simply by exposing the timber freely to the air in a dry place, sheltered, if possible, from sunshine and high winds. The seasoning yard should be paved and well drained, and the timber supported on stone or cast-iron bearers, and piled so as to admit of the free circulation of air over all the surfaces of the pieces.

Natural seasoning to fit timber for carpenters' work usually occupies about two years; for joiners' work, about four years; but much longer periods are sometimes employed.

To steep timber in water for a fortnight after felling it extracts part of the sap, and makes the drying process more rapid.

The best method of *Artificial Seasoning* is that of which the principle was first proposed by Sir Samuel Bentham, and which consists in exposing the timber in a chamber or oven to a current of hot air. In Mr. Davison's modification of that method, the current of air is impelled by a fan at the rate of about 100 feet per second; and the fan, air-passages, and chamber are so proportioned, that one-third of the volume of air in the chamber is blown through it per minute. The best temperature for the hot air varies with the kind and dimensions of the timber; thus, for—

Oak, of any dimensions, the temperature should not exceed.....	105° Fahr.
Leaf-woods in general, in logs or large pieces.....	90° to 100°
Pine-woods, in thick pieces.....	120°
" in thin boards.....	180° to 200°
Bay mahogany, in boards one inch thick.....	280° to 300°

The time required for drying is stated to be as follows:—

Thickness in inches.....	1, 2, 3, 4, 6, 8;
Time in weeks.....	1, 2, 3, 4, 7, 10,

the current of hot air being kept up for *twelve hours per day* only.

The drying of timber by hot air from a furnace has also been practised successfully by Mr. James Robert Napier, in a brick chamber, through which a current is produced by the draught of a chimney. The equable distribution of the hot air amongst the pieces of timber is insured by introducing the hot air close to the roof of the chamber, and drawing it off through holes in the floor into an underground flue. The hot air on entering, being more rare than that already in the chamber, which is partially cooled, spreads into a thin stratum close under the roof, and gradually descends amongst the pieces of wood to the floor. The air is introduced at the temperature of 240° Fahr. The expenditure of fuel is at the rate of 1 lb. of coke for every 3 lbs. of moisture evaporated.

Many experiments have been made on the loss of weight and shrinkage of dimensions undergone by timber in seasoning; of which the details may be found in the works of Fincham on "Shipbuilding," Tredgold on "Carpentry," Mr. Murray on "Timber," &c. The results of those experiments vary so much

that it is almost impossible to condense them into any general statement. The following shows the limits within which they generally lie:—

Timber.	Loss of Weight per Cent.	Transverse Shrink- ing per Cent.
Red Pine.....	from 12 to 25 2½ to 3
American Yellow Pine.....	" 18 to 27 2 to 3
Larch.....	" 6 to 25 2 to 3
Oak (British).....	" 16 to 30 about 8
Elm, ".....	" about 40 about 4
Mahogany.....	" 16 to 25	

24. *Durability and Decay of Timber.*—All kinds of timber are most lasting when kept constantly dry, and at the same time freely ventilated.

Timber kept constantly wet is softened and weakened; but it does not necessarily decay. Various kinds of timber, some of which have been already mentioned, such as greenheart, elm, and beech, possess great durability in that condition.

The situation which is least favourable to the duration of timber is that of alternate wetness and dryness, or of a slight degree of moisture, especially if accompanied by heat and confined air. For pieces of carpentry, therefore, which are to be exposed to these causes of decay, such as the planking of a ship's side, the stem and sternpost, the timbers of the hold, &c., the most durable kinds of timber only are to be employed, and proper precautions are to be taken for their preservation.

Timber exposed to confined air alone, without the presence of any considerable quantity of moisture, decays by "*dry rot*," which is accompanied by the growth of a fungus, and finally converts the wood into a fine powder.

Table A, in the Appendix to the Third Division, shows the comparative durability of timber, for purposes of shipbuilding, as estimated by the committee of Lloyd's. Table H (also extracted from Lloyd's Register, by permission of the committee) gives the same information in a more detailed form, and arranged in a different way.

25. *Preservation of Timber.*—Amongst the most efficient means of preserving timber, are good seasoning and the free circulation of air.

Protection against moisture is afforded by oil-paint, provided that the timber is perfectly dry when first painted, and that the paint is renewed from time to time. A coating of pitch or tar may be used for the same purpose.

Protection against the dry rot may be obtained by saturating the timber with solutions of particular metallic salts. For this purpose Chapman employed copperas (*sulphate of iron*); Mr. Kyan, corrosive sublimate (*bichloride of mercury*); Sir William Burnett, *chloride of zinc*. All these salts preserve the timber so long as they remain in its pores; but it would seem that they are gradually removed by the long-continued action of water.

Dr. Boucherie employs a solution of *sulphate of copper* in about one hundred times its weight of water. The solution, being contained in a tank about 30 or 40 feet above the level of the log, descends through a flexible tube to a cap fixed on one end of the log, whence it is forced by the pressure of the column of fluid above it through the tubes of the vascular tissue, driving out the sap before it at the other end of the log, until the tubes are cleared of sap and filled with the solution instead.

Timber is protected against wet rot, dry rot, and white ants, by Mr. Bethell's process of saturation with the liquid called

commercially "*creosote*," which is a kind of pitch oil. This is effected by first exhausting the air and moisture from the pores of the timber in an air-tight vessel, in which a partial vacuum is kept up for a few hours, and then forcing the creosote into those pores by a pressure of about 150 lbs. on the square inch, which is kept up for some days. The timber absorbs from a *ninth* to a *twelfth* of its weight of the oil.

26. *Strength of Timber*.—Amongst different specimens of timber of the same species, those which are most dense in the dry state are in general also the strongest.

Tables of the results of the most trustworthy experiments on the strength of different kinds of timber strained in various ways have been given at the end of the Third Division.

The following are some general remarks as to the different ways in which the strength of timber is exerted:—

The *Tenacity along the grain*, depending, as it does, on the tenacity of the fibres of the vascular tissue, is on the whole greatest in those kinds and pieces of wood in which those fibres are straightest and most distinctly marked. It is not materially affected by temporary wetness of the timber, but is diminished by long-continued saturation with water, and by steaming and boiling.

The *tenacity across the grain*, depending chiefly on the lateral adhesion of the fibres, is always considerably less than the tenacity along the grain, and is diminished by wetness and increased by dryness. Very few exact experiments have been made upon it. Its smallness in pine-wood as compared with leaf-wood forms a marked distinction between those two classes of timber,

the proportion which it bears to the tenacity along the grain having been found to be, by some experiments,

In pine-wood, from 1-20th to 1-10th.

In leaf-wood, from 1-6th to 1-4th, and upwards.

II. The *Resistance to Shearing*, by sliding of the fibres on each other, is the same, or nearly the same, with the tenacity across the grain.

III. The *Resistance to Crushing* along the grain, depending, as it does, on the resistance of the fibres to being crippled or "upset," and split asunder, is greatest when their lateral adhesion is greatest, and was found by Mr. Hodgkinson to be nearly twice as great for dry timber as for the same timber in the green state. In most kinds of timber, when dry, it ranges from one-half to two-thirds of the tenacity.

Experiments have been made on the crushing of timber across the grain, which takes place by a sort of shearing; but they have not led to any precise result, except that timber in general is both more compressible and weaker against a transverse than against a longitudinal pressure; and consequently, that intense transverse compression of pieces of timber ought to be avoided. Certain special kinds of timber are valued for the property of resisting compression across the grain well. Of these the most generally used is *Lignum Vitæ*, already mentioned in Article 19; to which may be added boxwood, iron-wood, and ebony.

IV. The *Modulus of Rupture* of timber, which expresses its resistance to cross-breaking, is usually somewhat less than its tenacity, but seldom much less.

CHAPTER II.

DESCRIPTION OF THE PARTS OF A SHIP.

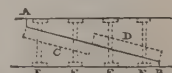
27. *Object of this Chapter*.—In Article 74 of the Third Division, the principal parts of a ship have been classed according to the manner in which they contribute to her strength. The object of the present Chapter is to describe those parts, together with the parts subordinate to them, more in detail, and with reference to the manner in which they are connected with each other. The processes by which those parts are made and put together will be described in the Third and Fourth Chapters of the present Division.

28. *Keel and its appendages*.—The *keel*, in *iron ships*, has various forms, which have been illustrated in Plate $\frac{H}{7}$. When it is a plain bar, the lengths of which it consists are either welded or scarfed together, the plane of each scarf being vertical. When it is built of various pieces, such as plates and angle-irons, they are made to break joint, as specified in the rules quoted in the Appendix to the Third Division. Some iron ships are built without a keel.

In *wooden and composite ships* the keel is a rectangular piece of timber, and usually of equal, or nearly equal, siding and moulding. In large vessels the siding of the keel for about a sixth or an eighth of its length at each end is often tapered at the rate of $\frac{1}{8}$ inch, or thereabouts, at each side, in the foot of length. The

lengths of timber of which the keel consists are scarfed together either with horizontal or with "up and down," or vertical *scarfs*, such as that represented in Fig. 1. The vertical scarf is the stronger. The length of the scarf is at least three times the "moulding" or depth of the keel. A and B are the *lips* of the scarf. C and D are *raised coaks*, *tablings*, or *tenons*, upon the thin ends of the two pieces, each fitting into a *sunk coak*, or mortise, in the opposite piece. The length of each coak is half the length of the scarf; the breadth, one-third of the moulding of the keel. E, E, E, E, are bolts to hold the scarf together.

Fig. 1.



Wooden ships are sometimes built upon a *temporary keel* of inferior timber, to save the permanent keel from the risk of decay. The temporary keel is removed piece by piece, and the permanent keel fitted in its place, after the framework has been built, and before the planking next the keel is permanently fastened.

A wooden keel has in each side a triangular *rabbet*, or groove, to receive the edge of the planking. This has already been frequently referred to in the Second Division.

To give a wooden keel a better hold of the water, its depth is increased by adding a *false keel* below it, of the same siding with the main keel. This should be so fastened that it may be knocked

off without injury to the main keel. For the same purpose, both wooden and iron ships are sometimes furnished with *bulge keels*, for an example of which in an iron ship (H.M.S. *Warrior*) see Plate $\frac{N}{3}$.

The depth of a wooden keel is sometimes increased at the top, by adding to it a piece of *dead-wood*, of the same siding with the keel, and of depth sufficient to admit of the floor-timbers being notched or *scored* upon it. The scarfs of this dead-wood should *shift* or break joint with those of the keel.

In wooden and composite ships as at present built, the keel often curves slightly upwards at its forward end, where it is scarfed to the stem, as will be further described in the next Article.

The main *keelson* lies directly above and parallel to the keel. In *iron ships* it is sometimes above and sometimes on a level with the floors, and has various forms, illustrated in Plate $\frac{N}{2}$. In *wooden ships* it always lies above the floor timbers, and is rectangular, and usually square, and nearly of the same scantling with the keel. It curves slightly upwards at both ends.

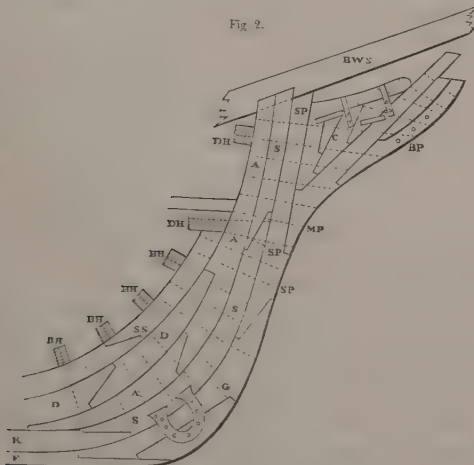
29. *Stem and its appendages*.—The *stem* forms a continuation of the keel, of the same material, curving upwards, and forming the extreme forward end of the ship. In *wooden and composite ships* it is scarfed, and in *iron ships* scarfed or welded, to the keel.

The upper end of the stem in wooden ships supports the bowsprit.

An iron stem is usually a flat bar of uniform, or nearly uniform, siding. It has sometimes been made with a rabbet at each side, to receive the foremost edge, or "hooding ends," of the outside plating; but this is now rarely practised.

A wooden stem has at its lower end a siding equal to that of the keel; and it is usual to make the siding enlarge gradually upwards, until at the upper end it is about one-third part greater than at the lower end: the object of this is to give a broad enough bed for the bowsprit. It always has a rabbet in each side (already referred to in the Second Division), to receive the foremost edge or hooding ends of the planking. The pieces of a wooden stem are scarfed to each other, and the lower piece to the keel.

The construction of the stem in wooden ships, and of various



pieces which form additions to its moulding fore and aft, are illustrated by Fig. 2.

K is the foremost end of the keel, curving slightly upwards, and

scarfed to S, the stem; A, the *apron*, being an addition to the moulding of the stem at its after side; D, the *dead-wood*, consisting of pieces built up so as to fill the space between the planking and timbers of the sides of the bow, wherever that space is not wider than the siding of the keel.

SS, the *stemson*, being a continuation of the keelson forward and upward, as the stem is a continuation of the keel.

D H, *deck-hooks*, being thwartship frames crossing the apron in a nearly horizontal position, to strengthen the bow and support the forward ends of the decks.

B H, *breast-hooks*, to strengthen the bow. These are made either of wood or of iron.

The following pieces compose the *head*, or *knee of the head*, being an addition to the moulding of the stem at its forward side, forming a beak-shaped projection, and serving partly as an ornament, and partly for securing the bowsprit and its rigging: S P, the *stem-piece* or *independent piece*; M P, the *main* or *lace piece*; B P, the *bobstay-piece*; C, *chocks*, or blocks to fill up the required shape of the head. The two oblong holes are for the *gammonings* to lash the bowsprit, and the round holes in the bobstay-piece for the *bobstays*; these are parts of the rigging to be described in the Fifth Division.

B W S is the bowsprit.

The keel, stem, apron, dead-wood, stemson, and the pieces of the head, are so arranged that their scarfs *shift*, or break joint; and, after being coaked, are bolted together by bolts arranged in the manner illustrated by the dotted lines in the figure; so that all the pieces of timber may act as far as possible like one piece.

The lower side pieces of the head are called *cheeks*, and the upper, *head-rails*. See for example, Plate $\frac{C}{1}$.

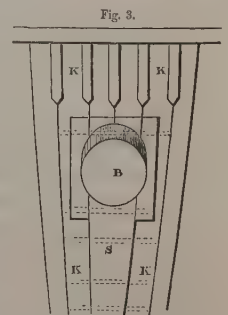
To give the bow of the vessel a good hold of the water, and so to promote her weatherliness, the *gripe*, G, is fastened upon the lower part of the stem by the aid of a pair of pieces of metal called *horse-shoes*, shown in the figure. The gripe forms a sort of continuation forward of the false keel F, and like it, should be so fastened that it may be knocked off without damaging the stem.

It is favourable to speed to have the forward edges of the stem, stem-piece, and gripe, *bearded*, or bevelled, so as to form a continuation of the curved surface of the entrance of the ship.

30. *Knight-heads and Hawse-pieces*.—In a *wooden ship* the bowsprit is secured against moving sideways, where it lies on the top of the stem, by means of a pair of pieces of timber called *knight-heads* (marked K, K, in the front view, Fig. 3), which stand close to, and one at each side of, the stem and apron, inside the outside planking. The *bowsprit-hole*, B, has its bottom formed by the top of the stem, S, and apron; its sides by the knight-heads; and its top by blocks of wood called *bowsprit-chocks*, filling the space between the tops of the knight-heads. Round the rim of the bowsprit-hole is a square or oblong projection called the *boxing*, left on the knight-heads and bowsprit-chocks, of depth equal to the thickness of, and forming the outside planking.

In an *iron ship* the bowsprit-hole is a tube, supported at the ends by two thwartship bulkheads.

In a *wooden ship* the *hawse-pieces* are the cant frames standing



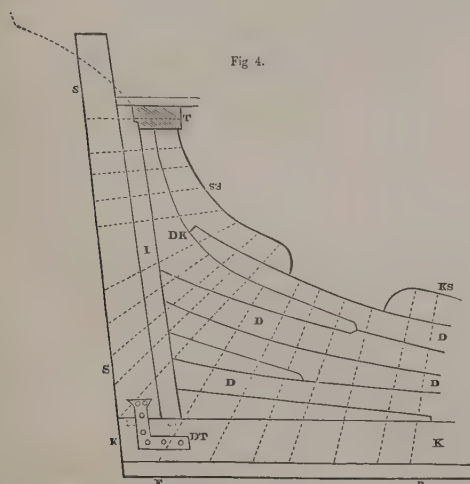
next to the knight-heads, to the number of six or seven, or thereabouts, at each side; they are fitted close together, so as to form a solid mass of timber for the passage of the four hawse-holes.

In an *iron ship* the hawse-holes are cast-iron tubes, having a strong projecting rounded rim inside and outside.

31. *Stern-post and its appendages.*—The *stern-post* is a straight piece, vertical or slightly raking, which rises from the after end of the keel.

In an *iron ship* the stern-post and after end of the keel are forged in one piece, which is scarfed or welded to the next piece of the keel. In a screw-steamer the forward or main stern-post or *propeller-post*, the after stern-post or *rudder-post*, the after end of the keel, and the *connecting piece* between the tops of the two stern-posts (forming together the frame of the *screw-port*), should all be forged in one piece (see Plates of H.M.S. *Warrior*, $\frac{B}{1}$, &c.): the main stern-post (as already explained in the Third Division and its Appendix) is made about twice as strong as in other vessels of the same size; and a ring is forged in it at a suitable height, for the passage of the propeller-shaft. When the screw is made to lift out of the water, the connecting piece forms a ring of such dimensions as to allow the screw, with its blades standing erect, to be lifted through it.

The stern-post (S, Fig. 4) of a *wooden ship* stands upon the



keel, K, into which it is mortised by means of two tenons, each of about one-third of the siding and one-fifth of the moulding of the stern-post. The keel and stern-post are further connected by means of a pair of metal dovetail plates, D T. Immediately before the stern-post is the *inner stern-post*, I, of the same siding with the stern-post. The *dead-wood*, D, is built up so as to fill the space between the outside planking and timbers of the two sides of the vessel, wherever that space is not wider than the siding of the keel. The piece, D K, is a *dead-wood knee*, sometimes used to connect the dead-wood with the stern-post. The *sternson*, S S, bears the same relation to the stern-post that the stemson bears to the stem; it is sometimes continuous with the keelson, K S, and sometimes, as in the figure, there is a gap between. T is a *deck transom* supporting the after end of a deck, in the same manner that a deck-hook supports the forward end. F is the false keel.

The butts of the pieces composing the stern-post, inner stern-post, dead-wood, and sternson, are made to shift or break joint;

and all those pieces are connected together by coaks and bolts, the bolts being arranged in a manner that is illustrated by the dotted lines in the figure.

It is advantageous to speed and to steering that the after edges of the stern-post should be *bearded*, so as to form a continuation of the curved surface of the vessel's run.

The *screw-trunk* and *screw-alley* of screw-steamer will be described in the Sixth Division.

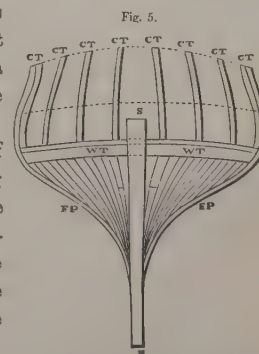
32. *Stern-framing.*—The sterns of ships have a variety of figures, depending more on taste and fashion than on principle or practical utility. The figures which most commonly occur are the *square stern*, being the oldest form, and still common in wooden merchant ships; and the *round stern* and *elliptical stern*, used in iron merchant ships, and in iron and wooden ships of war. The *counter* of the ship is that part which overhangs the stern-post. The head of the stern-post rises into the interior of the ship a short distance above the counter; and immediately abaft it is the *rudder-port*, being a hole in the counter for the passage of the rudder-head. The rudder-head, and the rudder-port, which it nearly fits, are cylindrical, the axis of the cylinder being the axis of motion of the rudder, and in one straight line with the axis of the pintles and braces by which the rudder is hinged to the stern-post; and the head of the stern-post is formed so as to admit of the rudder-head having a diameter sufficient for strength, agreeably to the principles of the Third Division, Article 86. (For an example of a stern-post so shaped at the head, see Plate $\frac{C}{7}$.)

In an *iron ship* the stern-framing may be constructed in the following manner:—The stern-post, at its head, may branch out into two ribs, forming a frame, like the other frames of the vessel. It is advisable that this frame should be either wholly or partially filled by a plate-iron bulkhead, with such openings in it as may be necessary for the passage of the tiller, or for other purposes. Then from that frame may spring *counter-frames*, each rising in a longitudinal or slightly oblique plane, having the same room and space, or nearly so, with the frames of the ship's body, and projecting out astern in such a figure as the designer may choose. For examples, see Plates $\frac{A}{2}$, $\frac{A}{3}$, and $\frac{F}{1}$.

In a *wooden ship* the stern-framing sometimes begins at a pair of cant frames, called *fashion-pieces* (F P, F P, Fig. 5). In the square stern the *wing-transom*, W T, extends across between the fashion-pieces, crossing in front of the stern-post, S, near its head. The wing-transom is nearly straight, having only a slight "round aft;" and it has also a "round up," depending on the taste of the designer.

At the upper and after edge of the wing-transom is a projecting moulding called the *margin*, the under side of which forms a continuation of the after side of the rabbet of the stern-post, so that the hooding-ends of the planking of the *buttock* abut against it.

Abaft of the fashion-pieces, and below the wing-transom, the framing may consist either of a series of transoms bearing the same relation to the stern-post that breast-hooks do to the stem, or of a series of radiating cant frames (called *after-timbers*), stepped partly on the dead-wood, and partly on *stepping-pieces* bolted to the



sides of the inner stern-post. The latter is the construction shown in Fig. 5.

Above the wing-transom spring *counter-timbers*, C T, C T, of such shapes as the designer may think fit. Their moulding planes are usually made to incline slightly inwards; and it is considered conducive to an elegant appearance that their upper ends should converge towards one imaginary point. Between the counter-timbers are the stern windows, if any.

Along the upper ends of the counter-frames or counter-timbers runs the *taffrail*.

The framing of a round or an elliptical stern consists of cant frames in radiating planes, running the whole way up to the taffrail; and in iron ships this mode of framing is often used.

The counter-timbers are crossed inside by *deck-transoms*, to support the decks; and outside by pieces called *stools* and *cornices*, which in wooden ships of the line were made to support one or more *galleries* or projecting balconies, accessible through the stern windows.

33. *Frames*, or *Transverse Ribs*, when made of *iron*, are of an L-shaped, Z-shaped, T-shaped, I-shaped, or trough-shaped section, as exemplified in Plate $\frac{H}{7}$. One angle-iron is generally continuous for the whole length of the frame, the lengths of which it is made being welded together. The *floor*, or lowest part of each frame, has usually two flanges, connected by a vertical web, as in the examples given in the Third Division and its Appendix; but in some instances the flanges have been connected by means of diagonal bracing instead of a web. The depth of the floors throughout the midship portion of the length of the vessel is regulated by considerations of strength, as explained in the Third Division, Article 80; in the fine parts near the ends of the vessel, which in a wooden ship would be occupied by dead-wood, the floor-plates or webs are triangular, and rise to the height of the *cutting-down line*, already mentioned in the Second Division, Article 21A. (See the longitudinal sections of the *Persia*, Plate $\frac{A}{2}$, *Formby*, Plate $\frac{F}{7}$, &c.)

A *wooden frame* is an *assemblage* built of two layers of pieces of timber, side by side, and breaking joint with each other, so that each *abutment*, *butt*, or joint of one layer, may be opposite the middle of a piece of the other layer. The pieces are called *floors*, or *floor-timbers*, *cross-timbers*, *half-floors*, *first futtocks*, *second futtocks*, &c., *long* and *short top-timbers*, and *lengthening pieces*. The upper and lower ends of each piece are called its *head* and *heel*.

A *floor* is a piece that lies across and is bolted to the keel, and has a long arm and a short arm. A *cross-piece* lies across and is bolted to the keel, and has two short arms. *Half-floors* are a pair of pieces abutting against each other on the top of the keel, and each of the length of the long arm of a floor. The bottom part of a rib consists either of a pair of floors, having the short arm of one beside the long arm of the other, or of a pair of half-floors lying beside a cross-timber.

The remainder of the assemblage, at each side of the vessel, is thus made up: one layer springs from the cross-timber head, or from the head of the short arm of a floor, as the case may be, and consists of the odd futtocks and long top-timber; the other layer springs from a half-floor head, or from the head of the long arm of a floor, as the case may be, and consists of the even futtocks and short top-timber. Sometimes the long top-timber is not long enough to reach to the head of the short top-timber; and then the required length is made up by a lengthening piece.

The upper surface of the floors and cross-timbers coincides with the *cutting-down line* already mentioned. The keelson lies on the top of the floors and cross-timbers, and is bolted through them to the keel.

In Fig. 6, K is the keel, FK the false keel, D dead-wood, KS the keelson, CTH a cross-timber head, H F H a half-floor-head, 1 F H the first futtock-head, 2 F H the second futtock-head, &c., T T the top-timber heads, PS the *planksheer* (or *gunwale*, as the case may be), mortised on to the top-timber heads, and projecting outside and inside so as to cover and sometimes slightly overhang both the outer and the inner skin. When a pair of long and short armed floors are used instead of a cross-timber and a pair of half-floors, CTH may be taken to represent the head of a short arm, and H F H that of a long arm.

The timbers of a frame are connected lengthwise at the timber-heads in two different ways: by *coaks*, or small cylindrical blocks of hard wood, fitting into holes in the heads and heels of the

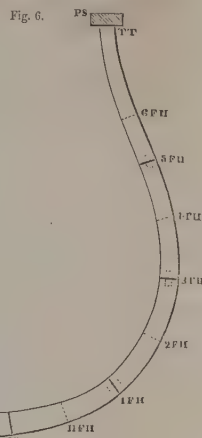


Fig. 7.

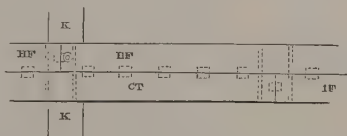


Fig. 8.



timbers, as shown by dotted lines in Fig. 6 and Fig. 7; or by *chocks*, of an obtuse wedge shape, scarfed to the heads and heels of the timbers, as shown at C in Fig. 8.

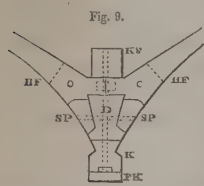
The timbers of a frame are connected together sideways by bolts near the head and heel of each timber, and also by coaks, spaced to about $1\frac{1}{2}$ times the moulding of the frame. The positions of such bolts and coaks are exemplified in the horizontal plan, Fig. 7, where K K is the keel; C T, a cross-timber; H F, H F, half-floors; and 1 F, a first futtock.

In the lower part of each frame, from the keel to the floor-heads, and sometimes as far as the first or second futtock-heads, the two layers of timbers of which the frame consists usually stand in close contact, forming a seam, which is caulked. Higher up, there is often a narrow opening for the circulation of air; the timbers being kept at the proper distance asunder by the coaks, while they are held together by the bolts.

When the spaces between the timbers of the hold are left open, *timber-holes* are bored through the frames at the floor and bilges, for the passage of water; but in H.M. service, and often in merchant ships also, the spaces between the floors and futtocks are filled in solid with fillings, and the seams are caulked.*

* From the calculations in the Third Division, Article 80, it is evident that the depths adopted in ordinary practice for frames, whether of iron or of wood, are barely sufficient for strength. This is probably one of the remaining effects of the old tonnage law, according to which the tonnage of vessels was

34. *Rising Floors—Cant Frames.*—The lower part of a *cant frame*, or of a square frame which has a sharp rise of floor, consists of a pair of half-floors springing either from a pair of rabbets in the dead-wood, or from a pair of *stepping-pieces* bolted to the sides of the dead-wood: which half-floors are connected together



across the top of the dead-wood by means of a *cross-piece* of suitable shape and dimensions. For example, in the cross-section, Fig. 9, K is the keel; FK the false keel; D the dead-wood; SP, SP, stepping-pieces; HF, HF, half-floors; K S the keelson; and the cross-

piece is shown by dotted lines.

35. *Longitudinal pieces in the Hold.*—The pieces used in the hold of an *iron ship*, to give additional longitudinal strength, such as *side or sister keelsons*, *hold stringers*, and *longitudinal frames*, have been sufficiently described and exemplified in the Third Division and its Appendix. When such pieces are fitted between the ribs instead of being above and inside them, they are called *intercostal keelsons*, *stringers*, &c.; and in such cases care must be taken to give them sufficient longitudinal connection across the ribs.

In *wooden ships* such longitudinal pieces always lie above and inside the frames, through which they are fastened to the outside skin. *Sister keelsons* usually run above the cross-timber or short-floor heads, and are made of lengths scarfed together like those of the main keelson. Additional longitudinal strength at the long-floor or half-floor heads and at the first futtock heads, is usually given by means of *thick strakes* in the inside planking, from two to four in number at each abutment.

36. The *Outside Skin* is bounded below by the keel, above by the planksheer, or by the gunwale, and at the ends by the stem, and by the stern-post and margin, or counter-rail.

The construction of the outside skin of *iron ships* has been described and illustrated in the Third Division and its Appendix; the precautions to be observed in fitting it on and fastening it, as well as that of wooden ships, will be described in Chapter IV. of this Division.

The usual thicknesses of the *outside planking* of wooden ships have been given in the Appendix to the Third Division, and the fitness of different kinds of timber for it has been discussed in that Division, and in Chapter I. of the present Division.

The outer edges of the planking fit into the rabbets of the keel, stem, stern-post, and margin, or counter-rail. The endmost

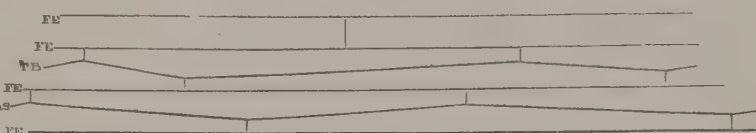
planks of the complete strakes are called *hoods*, and their ends, *hooding ends*. The endmost plank of a strake which stops short before reaching the stem or stern, is called a *stealer*.

The ordinary rules as to the *shift of butts* and *fastenings* of the planking, have been given in the Appendix to the Third Division.

When the *seam* between two strakes of plank forms a continuous line, it is said to be *fair-edged* or *straight-edged*. Every alternate seam is always fair-edged (FE, Fig. 10). The intermediate seams may be either fair-edged, or of a zig-zag shape, of which there are two kinds—*top and butt*, and *anchor-stock*.

Top and Butt planking (TB, Fig. 10) is when each plank has one edge fair, and the other consisting of a short slope and a long slope; the short slope being towards the butt, and the long slope towards the top, of the tree from which the planks are sawn. This

Fig. 10.



promotes economy of timber, especially when the material is British Oak.

Anchor-stock planking (AS, Fig. 10) is where each plank has one edge fair, and the other consisting of two equal slopes, so that the greatest breadth of the plank is at the middle. This is used where great strength is required, and where there are two strakes only, as in spirketting.

Diagonal planking has been mentioned in the Third Division, Article 79, and illustrated in Plate 9.

37. *Divisions of the Outside Skin.*—The following divisions of the outside skin apply both to wood and to iron:—

The *black-strake* is the strake next below the lower or gun-deck ports: the four strakes of planking, or the corresponding breadth of plating, immediately below the black-strake are called the *main wales*, and the strakes between the main wales and bottom planking are called the *diminishing stuff*, in consequence of their diminishing gradually in their thickness from that of the main wales to that of the bottom planking.

The strakes between the gun-deck ports and the middle-deck ports, in three-decked ships of war, are called the *middle wales*; and those between the middle-deck ports and the main-deck ports are called the *channel wales*.

In two-decked ships of war the strakes between the lower or gun-deck ports and main-deck ports are called *channel wales*.

The strakes above the main-deck ports extending up to the planksheer are called *sheer-strakes* in all classes of ships.

In merchant ships, the term *wales* is applied to the whole outer skin of the side below the sheer-strakes.

The planking between the ports fore and aft is called *short stuff*.

The *garboard strakes* are the two strakes next the keel on each side, and are made thicker than the rest of the bottom. In wooden ships, they are sometimes made of a thickness nearly equal to two-thirds of the depth of the keel, through which they are bolted to each other (a practice introduced by Mr. Lang); and the adjoining strakes of the bottom, to the number of four or five, have a gradually diminishing thickness, and rabbetted seams. (See Fig. 11, in which K is the keel, FK the false keel, and GS, GS, the garboard strakes).

computed from the outside measurement; so that it became an object to the owners that the outside measurement should exceed the capacity of the ship as little as possible.

In iron vessels the deficient strength of ordinary frames is made up for by the use of bulkheads, partial bulkheads, and longitudinal ribs, as explained in the Third Division, Articles 80, 81, and 83.

Now that the registered tonnage of vessels is computed from their real internal capacity, the same motive no longer exists for keeping the outside measurement as small as possible; and Mr. Peter Christie of Greenock, availing himself of that fact, has proposed a method of constructing wooden ships which (besides possessing other advantages) is calculated greatly to increase the transverse strength of a vessel of a given capacity, without increasing the expenditure of timber in her frames. Each frame-timber is to have its depth or moulding increased (say to double the present depth), and its breadth or siding diminished (say to half the present breadth), so that its sectional area may be unchanged. The additional moulding is to be added at the outside, so as to increase the outside breadth of the ship, and not to diminish her capacity. There is to be (at all events in the upper-deck) a beam to each frame, of the same siding with the frame, and connected at each end with the frame by means of a pair of flat plate-iron knees, so that the beam and frame may form one rigid hoop. As the narrowness of the frame-timbers will make the use of trenails unadvisable, the fastenings of the planking are to be metal bolts, passing through cylindrical coaks. The difficulty and expense of getting a sufficient quantity of timber deep enough for the frames of very large vessels, may be met by building such frames of two layers in depth as well as in breadth; such layers being made to break joint, and bolted and coaked together.

The *paddle-box beams*, and the parts which they support, will be treated of in the Sixth Division.

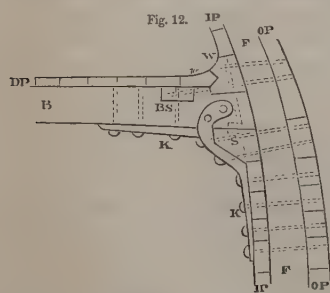
Where a mast passes through a deck, the hole for the mast is surrounded by a framing called the *mast-partners*, made of iron or of wood, and consisting of pieces of scantling equal or nearly equal to that of the beams. The two principal pieces lie fore and aft, with the mast between them: they are fastened to and supported by the two nearest beams.

The *flat of a deck* usually consists of planks lying fore and aft. Sometimes the planking of decks has been laid diagonally, in order to give transverse stiffness; but it is better to lay it fore and aft, for the sake of longitudinal strength, and to obtain transverse stiffness by means of flat *diagonal braces* of iron, laid above the beams and below the planking, as already described in the Appendix to the Third Division.

In iron ships, great additional longitudinal strength is gained by plating the upper deck with iron, over which planking is laid. The same object is promoted, whether in wooden or in iron ships, by *deck-stringers*, and by the *shelves* and *water-ways*, which run fore and aft along the ship's sides, the shelves below, and the water-ways above the beam-ends. In iron ships, the water-ways are usually made of plates lying flat on the beams; and they sometimes have a rising flange or ledge at the inner edge, of the depth of the planking, so as to form an open channel for water, and sometimes are covered by wooden water-ways (see Plate $\frac{11}{2}$). In wooden ships, the water-ways are strong pieces of timber, of a figure to be illustrated in the next Article.

Riding-bitts will be further described in the Fifth Chapter of this Division.

41. *Connection between Beams and Sides*.—The rigid connection of the beam-ends to the sides is of great importance both to general and local strength, as has been explained in the Third Division, Articles 80, 81, and 85. This is effected, in iron ships, by means of iron *knees* or of *bracket-ends* to the beams, and in wooden ships, by knees of wood or of iron, aided by the way in which the beams are fastened to the shelf-pieces and water-ways. The usual arrangement of those fastenings is illustrated by Fig. 12,



which represents a cross-section of part of the side of a wooden ship. F is a frame; OP the outside planking; IP the inside planking; B a deck-beam; DP the flat of the deck; S the shelf, to which the beam-end is coaked, the coak being marked by dots; W the thick water-way; w

the thin water-way, being of a gradually diminishing thickness from the thick water-way to the flat of the deck; BS the *binding strake* or *letting-down strake*, half-notched into the beams; K, a *forked iron knee*. This knee has four arms; one, called the *beam-arm*, is bolted to the under side of the beam; another, called the *side-arm*, to the ship's side, through the frame, and if necessary through a chock of such a shape and size as may be required to fill the space; the two remaining arms form a fork, embracing the sides of the beam, and bolted through it. The positions of bolts are marked by dotted lines.

42. *Bulwarks, Planksheers, Rails, &c.*—*Bulwarks* are those

parts of a ship's side which rise above the actually uppermost decks of a ship. The waist, quarter deck, and forecastle, or the flush deck, of all ships of war, and of some merchant ships, have close bulwarks. Their vertical framing, in wooden ships, consists of the top-timbers of the frames; in iron ships it may consist of the tops of the frames, or of strong wooden *bulwark-stanchions* secured to the tops of the frames and to the inside of the sheer-strake. The outside of the bulwarks may be either plated or planked, and the inside also if required. Along the top runs a piece, already mentioned in Article 33 of this Division, as well as in various preceding Articles, called the *gunwale* at the waist of a ship, and the *planksheer* at the quarter deck and forecastle of a ship with a waist, and throughout the whole length of a flush-decked ship. The upright or curved pieces of timber that connect the gunwale with the planksheer are called *drift pieces*. In some merchant ships the gunwale is continued all round, a little above the level of the quarter deck and forecastle (or spar deck as the case may be) and those decks have instead of bulwarks an open railing, with or without a netting. The poop, top-gallant-forecastle, and bridge deck, have almost always an open railing only. On the poop of wooden ships of war, that railing is called the *rough-tree rail*. The bulwarks may or may not contribute to the general strength of the ship, according to the strength and rigidity of their construction.

A railing or balustrade standing athwartships across a deck (for example, at the forward end of the quarter deck, or of the round-house), is called a *breastwork*; and the beam under it is called a *breast-beam*.

Above the gunwale of ships of war are fixed the *hammock nettings*, consisting of a row of forked upright iron stanchions, supporting either a netting or a wooden trough, in which the seamen stow their hammocks during the day.

43. *Riders* are pieces of framework inside the general framework of the ship, for the purpose of giving additional strength. Those at present used consist chiefly of *riding keelsons* above the main keelson, and of iron diagonal braces inside the frames of the hold of a wooden ship. For the purpose of diagonal bracing of the sides of wooden ships, *diagonal plates* are used, between the frames and the outside planking. Both those methods of bracing are described in the Appendix to the Third Division.

44. *Various openings in Ships*.—The *hawse-holes*, for the passage of the cables, have been referred to in Article 30. They are usually four in number (except in small vessels, which have two only), and are usually placed in range of the main deck, and between the cheeks of the head (see the side elevations of various ships, in the Plates). They pass through two thick pieces of wood or iron called the *bolsters of the hawse*, each bolted outside the plating or planking of the bows. Each hawse-hole is lined with a cast-iron *hawse-pipe*; the usual dimensions of which are nearly as follows:—

Inside diameter, from 9 to 10 times the diameter of the cable-iron:—
Thickness, $\frac{3}{4}$ of the diameter of the cable-iron.

The *hawse-holes* are stopped, when required, with *hawse-plugs*, to exclude water.

Scuppers are holes, lined with leaden, mixed metal, or iron pipes, for discharging water from the deck into the sea. Each of those pipes is sometimes made in two lengths, connected respectively with the inner and outer skin.

Gun-ports or *portholes* will be described in the Seventh Division.

Ports or openings in the bulwarks, or in the side between decks, suited for the passage of goods, are called *cargo-ports* or *gangways*. They have shutters, or lids, capable of being closed water-tight.

Scuttles, or *light ports*, are small windows for light and air, which in iron ships are usually round, and in wooden ships square or round. They are closed by panes of thick glass, set in strong metal frames, and if necessary, in stormy weather, by dead-lights, or water-tight shutters.

The name of *scuttles* is also applied to small hatches in the decks, and their covers are called *cap-scuttles*.

The *cabin windows* are less used in iron than in wooden ships. They are of various shapes more or less distorted, so as to suit the round-up of the decks and the rake of the counter-timbers. They are capable of being closed, in stormy weather, by water-tight outside shutters, called *dead-lights*. In wooden ships of war, they open upon overhanging balconies called *stern-galleries* and *quarter-galleries*.

45. *Bulkheads*, or partitions, when used to give transverse strength to the ship, and to divide her into water-tight compartments, have already been described in the Third Division and its Appendix. Bulkheads of lighter construction, and capable of being removed when required, are used to inclose cabins and other rooms, and to separate them from each other.

Longitudinal or *fore and aft bulkheads* are sometimes used to add to the longitudinal strength of a ship. They act like the web of a girder, to resist longitudinal racking and bending. For the same purpose, *longitudinal girders* are sometimes used in long shallow vessels (such as American River Steamers). These consist of an upper and lower stringer, connected together by a skeleton framework of upright and diagonal braces. The upper stringer is usually arched, and the lower stringer may form a keelson. Sometimes the duties of the upper stringer are done by wire *stay-ropes*, descending obliquely from the heads of upright masts to the ends of the vessel.

46. *Fillings between Frames*, in wooden ships, usually consist of small pieces of wood wedged into the spaces between the frames of the hold, and caulked; their object being to increase the water-tightness of the ship, and the resistance to compression, and to prevent the collection in those spaces of dirt, bilge-water, and vermin. For the same purpose in iron ships, and sometimes in wooden ships also, a mixture of Roman or Portland cement and clean sand, in the proportion of about two parts of cement to one of sand, is used sometimes alone, and sometimes along with bricks. Asphalte and various compositions are also used for filling those spaces. This subject has already been mentioned in the Appendix to the Third Division.

47. *Composite Ships*.—The principal objects of building composite ships are, to combine the durability and small friction of copper sheathing upon a wooden bottom with the greater strength, lightness, simplicity, and cheapness possessed by an iron frame. A principle requiring special attention in composite shipbuilding is, that every piece of iron must be completely *insulated*, or cut off, from electrical communication with any piece of copper or its alloys; otherwise the iron will be rapidly corroded.

In computing the strength of those parts of composite ships in which wood and iron lie side by side, and work together to resist the same force, regard must be had to the principles stated in the Third Division, at the end of Article 75, and

also to the facts that the expansion of wood by heat is insensible, while that of iron is about $\frac{1}{800}$ of its length for the difference of temperature between the freezing and boiling points of water.

Those facts and principles appear to lead to the conclusion, that it is better in composite shipbuilding to make all pieces which lie fore and aft of wood, and all those which lie athwartships or diagonally, or which stand upright, of iron, than to combine pieces of different materials in positions parallel to each other.

Many different ways of constructing composite ships have been invented. The following are the best known:—

A. *Wooden Skin and Iron Framework*.—The system of composite shipbuilding most generally practised is that patented by Mr. Jordan in 1849.* The whole outer skin, including keel, stem, stern-post, and planking, is of wood, arranged as in the skin of an ordinary wooden ship; and the framework inside the skin, including frames, beams, keelsons, stringers, shelf-pieces, water-ways, hooks, transoms, diagonal braces, &c., is of iron, arranged nearly as in an ordinary iron ship. It is to this system in particular that the Underwriters' Rules refer, which have been quoted in the Appendix to the Third Division.

The use of trough-shaped or "channel" iron for the frames (a form at once convenient and strong), has been introduced by Mr. Bettely.

The bolts which fasten the skin to the frame are of iron, generally "galvanized" or coated with zinc; and their outer ends are countersunk in holes of such a depth, that the iron bolts can be electrically insulated from the copper sheathing, by plugging the holes with pitch or other suitable non-conductor of electricity.

B. *Inner Skin Iron, Outer Skin Wood*.—In Mr. MacLaine's system of composite shipbuilding, there is a complete water-tight inner skin of iron plating, with its internal framing—viz., beams, stringers, keelsons, bulkheads (complete or partial), stanchions, platforms, &c.—of iron also. Outside the inner skin are rivetted transverse iron frames. Into each alternate space between the iron frames, is fitted a wooden frame, which is bolted to the two contiguous iron frames with fore-and-aft galvanized iron bolts. It projects nearly half its depth outside the iron frames. Outside the wooden frames is a complete wooden skin (including keel, stem, stern-post, and planking), fastened to the wooden frames with treenails or mixed metal bolts. The apron, inner-post, and dead-wood, are inserted between and bolted to angle-irons, which are rivetted upon the inner skin. The outside of the inner skin is coated with cement, pitch, or paint, before putting on the wooden frame and skin. The spaces between the wooden frames are ventilated artificially: in the armour-plated parts of ships of war, they may be filled up solid with wood. Any water which may get through the outer skin lodges in the spaces between the wooden floors, whence it is removed by pumping. To diminish, when required, the weight and cost of the wooden framing, every alternate wooden frame may be omitted, and replaced by a bent plank of teak or other sufficiently strong timber, fastened to the outer skin only.†

According to Mr. Hein's system, the ship has a complete inner skin of iron, on the outside of which are rivetted trans-

* This patent was prolonged for seven years in 1863.

† See "Improvement of Naval Construction," by Alexander MacLaine, Esq., Assoc. I.N.A.

verse iron frames of a Z-shaped section. The spaces between the frames are filled up tight with timbers shaped like the frame-timbers of a wooden vessel, rabbeted into each other, and bolted with fore-and-aft bolts to each other and to the iron ribs. Between those filling timbers and the iron skin is a layer of pitch. The filling timbers project beyond and completely cover the outer flanges of the iron ribs; and the seams between them are caulked, and have a groove running down the outside edge of each of them. Outside the filling timbers is a wooden skin, consisting of a keel and fore-and-aft planking laid and caulked in the usual way, and fastening to the filling timbers with mixed metal nails and bolts. Any water which penetrates through the seams of the outer skin is caught by the grooves along the seams of the filling timbers, runs down those grooves into a limber or water-passage cut in the upper side of the keel, and is drawn from that passage by a pump. The beams are of iron, connected with the inner skin by means of gussets or knees.

In this class might, perhaps, be included Mr. Grantham's method of sheathing iron ships with wood and copper; but it belongs more properly to the subject of the protection of ships' bottoms.

c. *Iron frames imbedded between two Wooden Skins.*—(System of Captain Macgregor Skinner, R.N.) In this system there are two complete wooden skins, each laid fore and aft; the outer

skin consisting of keel, stem, stern-post, and planking; and the inner skin of keelson, ceiling or inner planking (made of planks or logs as may be required for strength), shelf-pieces, water-ways, &c.; so that all longitudinal pieces in the ship are of timber. The transverse pieces are of iron (or other metal), and consist of ribs or frames, and deck beams. The two wooden skins have a layer of tarred felt between them, and are fastened directly together with copper or mixed metal bolts or wooden treenails, so that their seams break joint with each other; and they hold the iron ribs between them imbedded in grooves in the inner skin; thus those ribs are not weakened by rivet-holes, and are not in contact with electro-positive metal. Both skins are caulked, the inner one both inside and out; and the strakes of the inner skin (including keelson, shelf-pieces, &c.) are coaked and bolted to each other edgewise.

d. *Wooden Bottom and Iron Topsides.*—(Mr. Feather's system). In this system both frame and skin, up to the highest part that is at any time immersed, are built of wood, and both frame and skin of the upper part of the sides are built of iron. The iron frames terminate at their lower ends in broad forks or saddles, which sit upon and are fastened to the wooden parts of the sides.

On the subject of composite ships, reference may be made to a paper by Mr. Abegg, in the "Transactions of the Institution of Naval Architects for 1864."

CHAPTER III.

SHAPING AND TOOLS.

SECTION I.—SHAPING IRON FOR SHIPBUILDING.*

48. *Shaping the Frames* of iron vessels consists mainly in *cutting* or *shearing* the angle-iron bars to the proper length, *bending* them so as to give the proper figure to the moulding edge, and *beveling* them.

I. The *shearing* of angle iron is usually performed by a machine, consisting of a fixed and a moveable cutter: the fixed cutter is of the form of a right-angled triangular notch, in which the angle iron to be cut is laid with the angle downwards: the moveable cutter is a solid right-angled triangle, with the right angle pointing downwards; it is fixed in the lower end of a block which slides between vertical guides, and has a reciprocating motion given to it by an eccentric upon a rotating shaft, making twenty revolutions per minute, or thereabouts. The effort required to shear a piece of iron is about 50,000 lbs. per square inch of the area of the shorn surface; the work performed is about equal to that effort multiplied by half the thickness of the piece in the direction of shearing. For an equal area of steel, the effort is probably about double.

II. The *bending* of angle-iron frames to any sharp curvature is usually done while hot, upon a level platform, composed of large plates of cast iron called *levelling-blocks*, *levelling-slabs*, or *levelling-*

plates. These plates are completely covered with holes, about $1\frac{1}{2}$ inch in diameter, and 4 or 5 inches apart from centre to centre. The wooden mould of a given frame having been laid on the levelling-plates, the figure of the moulding edge is marked on them with chalk, and iron pins are stuck in the holes, so that when the iron rib is made to touch those pins it shall have the proper form. In order the more easily to produce any required figure, the heads of the pins are furnished with eccentric discs or cams,* by the shifting and turning of which the figure of the frame can be adjusted with great precision. Each disc has several centre holes, any one of which can be fitted on the pin. The iron bar for the frame having been raised to a bright orange heat in a reverberatory furnace, called a *reheating furnace*, is taken out by the smiths, laid on the levelling-plates, and rapidly bent, by means of tongs, hammers, mallets, and levers, so as to lie touching the heads of the pins.

Care should be taken that just enough of air for complete combustion, and no more, is admitted into the reheating furnaces; for any excess of air burns the iron. In fact, for the sake of safety against that evil, it is best that the supply of air should be slightly deficient, although the consequence may be that some smoke is given out.

III. *Cold bending* may be used when a slight uniform curva-

* For information on the subject of this Section, and on various other departments of the practice of Iron Shipbuilding, reference may be made to the well-known work of Mr. John Grantham.

* First introduced by Mr. J. R. Napier.

ture is to be given to a bar, or when a slightly bent bar is to be straightened. It may be done by means of a machine having a motion similar to that of the shearing machine already mentioned, and performed either in a vertical or in a horizontal plane. One side of the bar to be bent or straightened rests against a pair of fixed blocks with slightly rounded surfaces: midway between those fixed blocks, the opposite side of the bar is pressed upon by a block having a reciprocating motion. The position of the fixed blocks and the length of stroke of the moveable block are capable of adjustment, according to the alteration to be produced in the figure of the bars. After each stroke of the moveable block, the bar is pushed or dragged forward through a distance equal to about half the space between the fixed blocks.

IV. *Bevelling* of angle-iron frames, according to the bevellings given on the bevelling boards (Division II., Article 37), is performed by smiths while the iron is lying hot upon the levelling-plates, at the same time with the bending; and it is done by *opening* or *closing* the angle iron according as the bevelling forms an obtuse or an acute angle.*

The opening and closing of angle-iron frames requires careful superintendence; and special attention should be paid to the flatness of the outer or *faying* surface of the side-arm of the angle iron, to which the plating of the ship is to be rivetted; for if the opening or closing is carelessly done, that surface becomes concave or convex instead of flat, and the rivetting of the plating to it cannot be made secure. Care should be taken, also, that the frames are not split at the inside or outside of the angle, by the great strain that the process of opening or closing produces there.

49. *Shaping Plates* consists chiefly in cutting them to the required size and figure, planing their edges, and bending them. The figures and sizes of the plates required for a ship are shown on an expansion of her skin (Second Division, Article 38), on which their thicknesses are also written, or indicated by colours or shading (see Plate 4). The plates should be got from the manufacturer as nearly as practicable of the proper dimensions, so that there may be as little cutting needed as possible; but some cutting will always be required, especially near the ends of the vessel.

I. The *cutting* of plates is done by means of a shearing machine of sufficient size and power. The cast-steel cutters are both straight. The lower cutter is fixed and horizontal. The upper or moveable cutter has a slight slope, so that the shearing of a plate begins at one side and advances by degrees towards the other.

II. *Planing* the edges of plates is required at butt joints, in order that the fit may be accurate and close, for the sake both of water-tightness, and of the uniform distribution of compressive stress.

III. *Bending* plates, if the curvature is great, must be done after heating them in a reverberatory furnace of suitable dimensions; if the curvature is slight, it can be performed while they are cold, by the aid of a machine. The plate, in passing through

the plate-bending machine, rests upon and is carried forward by two horizontal rollers, with fixed bearings, and driven by suitable gearing. Between those rollers, the upper side of the plate is pressed upon by a free roller, the positions of both bearings of which are independently adjustable by screws, so as to give any required curvature to the plate, whether cylindrical or conical, constant or varying.

50. *Punching* and *drilling* are the means of making the holes through which the pieces of a ship's framework and skin are rivetted together. In either of those operations, two things have to be considered: first, the position and arrangement of the holes; and secondly, their figure, and the means of producing it.

I. The *position* and *arrangement* of the holes, and especially their *pitch*, or distance apart from centre to centre, must be exactly the same in two pieces that are to be rivetted together: the slightest want of accuracy in their correspondence with each other being fatal to good workmanship. The bad practice of stretching with a *drift* one or both of a pair of holes which do not truly correspond, so as to make a partial correction of the error, is never permitted in good shipbuilding.

The oldest and simplest way of making the rivet-holes in two pieces correspond, where the holes are punched or drilled one by one, is as follows:—the holes in one of the pieces having been made in the first place, the two pieces are laid together in their intended relative position; when plugs dipped in whitening, passed through the holes of the first piece, mark the spots on the second piece where the holes are to be made.

That process becomes unnecessary when machines are used which can punch or drill a whole row of holes of uniform pitch; so that when the two ends of the rows of holes in a pair of pieces correspond, all the intermediate holes correspond also.

II. *Figure of Holes*.—Drilled holes are cylindrical, being of the same diameter throughout; punched holes are conical, being of the diameter of the punch, at the side where it goes in, and of a somewhat greater diameter, at the opposite side, where the piece of metal from the hole drops out. The diameter of a punched hole, at the large end, is equal, or nearly so, to that of the hole in the *die*, or perforated plate of steel on which the plate or bar rests that is being punched. In order that the punching machine may work easily, the die must be at least from $\frac{1}{16}$ to $\frac{1}{8}$ of an inch larger in diameter than the punch; and by making the die wider still, holes of any degree of taper required in practice may be punched; but it is only the smaller end of a punched hole that is perfectly accurate in diameter and position; the larger end is apt to be somewhat irregular, and if it widens very much, to be more or less rough and ragged. In other respects the conical form of the punched holes is advantageous, as enabling the rivets to hold the plates more firmly.

Hence, where *three or more* layers of plates or bars are to be rivetted together, it is advisable, for the sake of accurate fitting, that the holes should be drilled, and not punched, except in the outermost layers; but where *two layers only* are to be rivetted together, punched holes will give a more firm fastening, provided *the small ends of the holes are placed together*. Hence the indispensable rule, that *all punched holes should be punched from the faying surface of the plate or bar*; that is, from the surface which is to touch that of the piece to which it is to be rivetted. In a piece with two faying surfaces, the holes should be drilled.

The outer ends of all rivets in the outside plating of an iron

* The opening and closing of angle iron is not only difficult to perform correctly, but it strains the material very severely at the angle, and is always more or less injurious to its strength. The Editor of this Treatise has suggested (in a paper read to the Institution of Naval Architects in 1865), that the required bevellings might be given much more easily, and without the slightest overstraining of the iron, by twisting the angle iron instead of opening or closing it. This would have the additional advantage of placing the thwartship arm of each frame normal to the skin, being precisely the position that is most favourable to strength and stiffness. The beam-ends, and the edges of bulkheads, where rivetted to the frames, would have to be slightly bent horizontally, and the floor-plates slightly twisted, so as to be normal to the skin also; and this too would be advantageous in point of strength. Mr. J. R. Napier has pointed out, that the twisting should be performed on the angle iron while straight, before bending it on the levelling-plates to the required curve; and that as the twist in most cases would be very slight, it might be given by a suitable machine to the bar when cold.

ship are countersunk; and for that purpose the holes in the outer plates must have a conical enlargement, which is some-

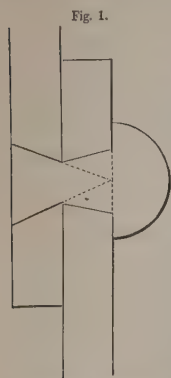


Fig. 1.

times drilled with a conical tool called a *countersink drill*, and sometimes made simply by punching the hole of a sufficiently spreading conical form. In either case the hole should have the same figure; that is to say, there should not be a mere conical countersink for a small depth inwards from the outer end of the hole, but *the entire hole should form one cone*: and in order that the rivet after its contraction in cooling may not only hold the plates together, but continue to fit the hole in the outside plate with equal tightness, *the apex of that cone should be in the plane of the inner surface*

of the innermost piece through which the rivet passes.*—See Figs. 1 and 2.

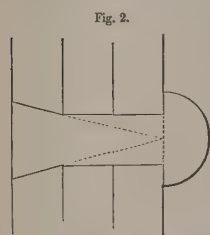


Fig. 2.

Plates and bars of more than one inch thick are almost always drilled, and not punched.

The force required to punch a hole in strong wrought iron is about 50,000 lbs. for each square inch of the fractured surface, as in the case of shearing; and the work done is nearly equal to that force multiplied by half the thick-

ness of the plate or bar.

SECTION II.—SHAPING TIMBER.†

51. By the *Conversion* of timber is meant, the cutting (in general with the saw) of logs of timber into pieces nearly of the shape required in shipbuilding.

Great experience, judgment, and care are necessary on the part of the *converter* who conducts this process, in order that he may, in the first place, select the logs best suited for given purposes, and then cut up those logs in the most efficient and economical way.

The following are the chief principles to be attended to in converting timber:—

I. Every piece is to be *as little grain-cut as possible*; in other words, the natural figure of the fibres of the wood should approach as near as possible to that of the principal pieces to be cut from it: thus, long and straight logs are to be used for keels, keelsons, &c., and for sawing into planks and thick-stuff; shorter straight logs for beams, stern-posts, &c.; more or less curved pieces for stems, futtocks, transoms, &c.; and the most crooked pieces for hooks and knees.

II. Besides cutting one or more principal pieces from a log in the best possible way, regard should be had to the economical conversion of the remainder of the log into smaller pieces, such as chocks, carlings, &c.

III. Regard should be had to the blemishes of the timber, in converting it (see Article 15 of this Division). If those are of a bad kind, and especially if they consist in decay, or in a tendency to decay, the blemished parts must be cut out and

rejected. Clefts and shakes of small extent, in a large piece of timber, may be prevented from injuring its strength much, if care be taken to convert it in such a manner, that the damaged places shall be near the middle of the depth of the piece.

IV. For thick-stuff and planking, unblemished and straight-grained timber alone should be used. In sawing timber with "silver grain," or large medullary rays (see Article 17 of this Division), into planks, care should be taken that the silver-grain does not quite run through the planks, otherwise they will be liable to be split by the fastenings.

V. As the top of a log is naturally more durable than the butt, the butts of pieces of timber should as much as possible be placed in those situations in the ship's framing which expose them least to causes of decay; for example, the butt of a log should be the heel rather than the head of the stern-post; because wood lasts longer when always wet than when alternately wet and dry.

52. *Steaming and Bending*.—In the scarcity of naturally crooked timber, pieces of wood may be bent artificially, having first been softened by steaming or boiling. On being again dried, while in the bent position, the timber recovers its hardness, and retains its curvature.

The bending apparatus may be varied very much in detail; but essentially it consists of two rounded cast-iron blocks pressing upon that side of the timber which is to be convex, near its ends, and one, two, three, or any required number of such blocks pressing against the intermediate parts of that side of the timber which is to be concave; the position of each block being adjusted to the required curvature by means of a powerful screw.

So far as the fibres of the wood are compressed while in the soft state, it is found that their strength is but little impaired; but so far as they are stretched to any material extent, they are permanently weakened. To prevent this evil, before the bending is commenced a piece of plate iron, BB (Fig. 3), of the same

Fig. 3.



breadth with the timber, and somewhat longer, is placed in contact with that side of the timber which is to be made convex; and to that plate are rivetted or bolted a pair of angle-iron abutments, CC, in close contact with the ends of the timber. When the timber thus guarded is bent, the iron plate and abutments prevent any appreciable stretching of its fibres, so that the whole bending takes place by compression; and the result is, that the bent timber is not sensibly weaker than a naturally bent piece of the same figure.

53. The *Forming and Trimming* of timbers consists in taking the converted pieces of wood, and shaping them exactly to the required figures, by first *siding* them, or giving them the correct breadths; secondly, *moulding* them, or giving them the correct outlines and depths; and thirdly, *bevelling* them, where that operation is required.

Pieces that lie or stand amidships, like the keel, stem, stern-post, dead-wood, keelson, &c., are sided by laying off the half-siding each way from a central plane. They are then moulded,

* This rule appears to have been first put in practice by Mr. Thomas Hoey of Glasgow. It has been demonstrated by various authors.

† On the subject of this Section reference may be made to Fincham's "Outline of Shipbuilding."

by being cut to the shape and dimensions of the moulds, furnished from the mould-loft (Second Division, Article 36), which show not only the outlines of the several pieces, but the forms and positions of the scarfs by which they are to be connected together.

The stem, with the dead-wood and other adjoining pieces, having been treenailed and coaked together, the stations for hooks and for cant frames are marked on them, and the rabbet for stepping the cant frames is cut. The same description applies to the dead-wood and other pieces adjoining the stern-post.

Pieces that have a moulding side, such as frame-timbers, are in the first place trimmed truly plane, or "out of winding," as it is called, on the moulding side; the siding is then laid off from that side; then the mould is laid upon the moulding side of the timber, and the moulding-edge drawn and cut to its true figure; then the *sirmarks*, or points of intersection of the moulding-edge with the riband-lines, and the other bevelling points, are marked; then the depths are laid off, so as to enable the inner edge of the moulding side to be drawn and cut; and lastly, the piece is trimmed to the true bevelling, which is shown for each sirmark, &c., on the bevelling-board (Division II., Article 37), and is transferred to the timber by the aid of the *bevel*, an instrument consisting of two straight-edged pieces hinged together—one, called the *stock*, which is applied to the moulding side, and at right angles to the moulding edge; and the other, called the *tongue*, which is set so as to form the proper angle of bevelling with the stock, and which touches the bevelling surface of the timber when it is correctly formed.

54. *Coaks*.—Where two pieces of timber are to be connected by a cylindrical *coak* or *dowel*, they are laid together in their proper relative position, and a hole is bored with a small auger through both pieces at once, to mark the centre of the coak; then a hole of the diameter and half-depth of the coak is bored from the faying surface of each piece, with a drilling tool of the proper size, in what is called a *dowel-engine*. The cylindrical coaks themselves are turned out of hard and durable wood.

A *mortise*, or *sunk coak*, is a hollow cut in a piece of timber, to fit a *tenon*, or *raised coak*, on another piece.* When cut by hand, mortises and tenons are usually rectangular; when a mortise is cut by machinery, it is often oblong, with semicircular ends, being made by means of a boring tool which has a traversing motion from side to side equal in extent to the length of the straight sides of the mortise. Square mortises, however, are sometimes cut by a machine tool, consisting of a hollow square chisel having an auger working through it, which removes the chips produced by the chisel along with those produced by its own action.

55. *Treenails* are turned out of hard, strong, and durable timber, such as the best oak and teak. The only woods suited for this purpose are those which have considerable strength across

as well as along the grain. The diameters of treenails range from 1 inch to $1\frac{3}{4}$ inch (see Table D of Lloyd's Rules).

Compressed treenails are made by a machine, in which, after having been softened by steaming, they are forced in at the larger end and out at the smaller end of a tapering steel tube, so as to reduce them to about two-thirds of their original diameter. Upon being again moistened, they gradually swell out to their former size. They make a very tight fastening, but are not to be used except in large and thick pieces of timber, lest by swelling they should split the pieces that they are used to connect.

The mode of using treenails belongs to the next Chapter.

56. *Shaping-machines for Wood*.—The most generally used machines for working in timber are *saws*, of different kinds. Those for making straight cuts, as where the timber is to be sawn into planks, may be either circular or reciprocating. The log to be sawn is supported by a moveable frame called a *carriage*, by the slow motion of which it is *fed* to the saw. The feed-motion to circular saws is usually horizontal; to reciprocating saws, either horizontal or vertical, according as the saw is vertical or horizontal. A vertical saw usually cuts during its down-stroke only, and the feed-motion accordingly takes place wholly during the down-stroke of the saw, and ceases during the up-stroke: a horizontal saw may cut both ways, and then the feed-motion is continuous; and the motion of the saw, which is straight, takes place upon guides slightly curved in a direction convex towards the advancing log, in order that the cutting action of *each tooth* of the saw may take place in a direction towards the centre of the log, and that at the same time the order in which *successive teeth* of the saw come into action may be from the centre to the outside. (Pentzlin's Saw.) Reciprocating machine saws should have the reactions due to their motion carefully counterpoised, so that there may be a "running balance," as well as a "standing balance;" otherwise the machinery, when working fast, will overstrain both itself and the building.

A machine saw for cutting timber into curved shapes, is usually an endless flexible *band* of steel, with saw teeth on its edge, carried and driven by pulleys, and passing downwards through a hole in a cast-iron table. A piece of wood, being placed on the table and properly guided, may have a saw-cut made in it of any required figure; and by tilting the table, or inclining the band-saw, or by both those motions combined, the cut may be bevelled in any required manner.

In *shaping-machines* for cutting long pieces of timber to cross-sections suited for planksheers, rails, thick water-ways, &c., the timber is usually fed horizontally by a carriage moving like that of a sawing machine, to a set of rapidly rotating discs armed with steel cutters resembling chisels and small adzes: the arrangement of the discs and cutters is capable of being varied, so as to produce different forms of cross-section.

* This is sometimes also called *tabling*.

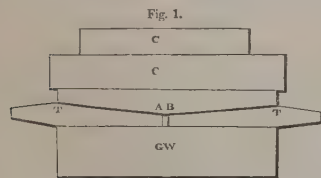
CHAPTER IV.

BUILDING.

57. *Building Slip—Blocks, &c.*—A ship is usually built upon a piece of ground called a *slip*, which in most cases slopes lengthwise towards the water. The slip should have a firm foundation, and, if possible, a floor of masonry, concrete, timber, or iron. Perfect solidity of the slip is essential to the strength and safety of the vessels built upon it.

To support the keel of the ship, a row of temporary *building-blocks* are built, 4 feet apart, or thereabouts, from centre to centre. Each block is built of several pieces of timber, one above another; the lowest and largest of them are called *groundways*: they are from 12 to 15 inches square, and unless the foundation is so firm as to make it unnecessary, they lie lengthwise, and form a platform of timber under the ship of 4 or 5 feet broad. The other pieces lie athwartships: they gradually diminish in size, and are more or fewer in number according to the height of the block; the uppermost are called *caps*.

With a view to the future launching of the vessel, the blocks are made capable of being removed from below her keel. Fig. 1



is a thwartship view of a block; G W, groundways; C, C, caps; A B, angle block, faced on the lower side with a thin plate of iron; T, T, wedges called *templates*, which rest on

the groundways and support the angle block. When the block is to be struck, or removed, the templates are knocked out, which removes the support from the angle block and the pieces above it. When angle blocks are not used, the removal of the blocks is effected by splitting the caps.

In order that the workmen may easily get at the bottom of the ship, the *lowest* block is usually made about *two feet* high: it is generally under the forefoot of the ship; the heights of the other blocks depend on the inclinations of the ship and of the keel.

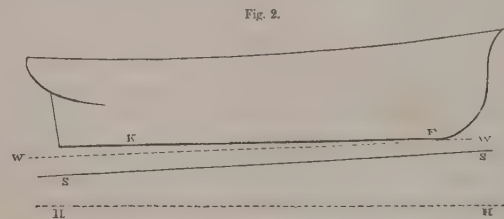
Three rates of inclination have to be attended to in building a ship—that of her keel, that of the intended *sliding-ways* on which she is to be launched, and that of the slip. The inclination of the ship's keel, when it is not intended to launch her on her keel, is a matter of choice and convenience; it is usually about 1 in 19 or 20, descending towards the water. The objects of making it descend towards the water are, to avoid excessive height in the building-blocks at the lower end of the slip; and to give the ship a bearing on the water as soon as possible when she is launched. Should it be intended to launch the ship on her keel, the inclination of the keel must be the same with that of the sliding-ways. The ordinary *inclinations of the sliding-ways* are as follows:—

For the smallest vessels, from.....	1 in 12 to 1 in 14
For average vessels, about.....	1 in 16
For the largest vessels, from.....	1 in 20 to 1 in 24

The inclination of the sliding-ways is made gradually flatter for larger vessels, to prevent them from acquiring an excessive speed when they are launched.

The inclination of the floor of the slip is very much a matter of convenience; but it is usual to make it steeper than the steepest inclination of the sliding-ways required for launching the smallest vessel that is to be built upon it; that is to say, about 1 in 9 or 10. This, however, is not absolutely necessary, provided care is taken to adjust the height of the blocks, so that the forefoot of the vessel in launching shall clear the lower end of the slip by about 9 inches.

The heights of the blocks are determined by construction on the sheer-plan of the ship, as shown in the sketch, Fig. 2; in



which H H represents a horizontal line, and K F the lower side of the false keel, at its intended inclination. Through F, the forefoot of the vessel, draw W W at the intended inclination of the sliding-ways; this will represent the line of motion of the forefoot in launching. Set off 2 feet vertically downwards from F, to represent the height of the foremost block, and through the spot thus found, draw S S at the inclination of the slip. If S S is nowhere nearer to W W than 9 inches, the forefoot will pass sufficiently clear of the ground in launching; if not, the height of the foremost block must be increased.

The heights from S S to K F at the stations of the several blocks will show the heights of those blocks; and in flat-floored ships care must be taken to give sufficient height to the midship blocks, to enable the work of that part of the bottom to be done easily and well.

The preceding explanations are given on the supposition that the slip is to be launched in the ordinary way, stern foremost; so that she is to be built with her head at the upper end of the slip. Should it be determined to launch her head foremost, she must be built with her stern at the upper end of the slip. Ships are sometimes launched "broadside on;" and then the inclination of the slip must run athwartships, and its floor must be level longitudinally. A ship may be built in a dock, and when complete, floated by the admission of water, instead of being launched; and then the floor of the dock may be level both ways.

Building-blocks of 4 or 5 feet high and upwards, are stayed in a fore and aft direction with oblique shores, to prevent them from tripping.

58. In *Setting-up the Framework*, whether wood or iron, the first operation is to lay the pieces of the ship's keel on the building-blocks, and scarf them together. (The use of a temporary keel of inferior timber in building wooden vessels, has already been mentioned in Article 28 of this Division.) The keel is prevented from shifting sideways on the building-blocks by means of pins called *nogs*, driven into the blocks at each side of it. The vertical scarfs of the keel are caulked, as will be afterwards more fully described.

The stem and stern-post are set up in their proper positions, in a truly vertical fore and aft plane; they are supported in that position with suitable shores or props, and scarfed to the keel.

In wooden ships, the dead-wood, and other pieces which stand in the fore-and-aft midship plane, are brought to their proper places, and scarfed, coaked, and bolted to the keel, stem, and stern-post.

In iron vessels, the garboard strakes are at once put on, and rivetted to the keel, in order to provide a steady support for the floors: in wooden vessels this is delayed until the rest of the outside skin is about to be put on.

The stations of the frames are marked on the keel or dead-wood, as the case may be. In wooden vessels, the dead-wood of the keel is *scored* or notched, to receive the floors and cross-timbers.

The *raising of the frames* in a wooden ship is commenced by fastening together each pair of floors, or of half-floors with their cross-timber, bringing them to their station, and *crossing* them—that is, placing them perpendicular to the keel, and at the same time adjusting the scoring-down of the dead-wood, and, if necessary, trimming the floor-timbers, so as to bring their upper surfaces exactly to the level of the *cutting-down line*, as furnished from the mould loft by means of a *cutting-down staff*, on which are marked the heights of the cutting-down line above the keel at the several frames. The pieces of each side frame, having been fastened together, are brought to their station, set up, and fastened to the floors. In an iron ship, whole frames are often raised as one piece.

Each frame is kept in shape by temporary timber braces; those which run horizontally being called *cross-spalls*, and those which run diagonally and vertically, *shores*. The cross-spalls are nearly in the positions to be afterwards occupied by the deck-beams.

Each frame is adjusted so as to be exactly perpendicular to the keel, by the aid of a plumb-line or of a level.

The frames are kept in their proper positions by temporary wooden supports of the following kind:—Along each of the *riband-lines* (already often mentioned) outside the frame, runs a long piece of fir timber, about 5 inches square, called a *riband* where it crosses the square frames, and a *harpin* where it crosses the cant frames; at the *sirmarks*, where the ribands and harpins cross the frames, they are attached to them with temporary iron bolts or screws. The ribands and harpins are propped from the outside with sloping timber *shores*, abutting, in an open slip, against stakes driven into the ground; and in a dock, against the masonry of its sides.

The main keelson is laid, and fastened through the floors to the keel, after the frames have been set up. In wooden ships the keelson is coaked to the floors; it is also coaked to the dead-wood of the bow and stern, by coaks at intervals about equal to the room and space; and it is bolted through the floors and dead-

wood to the keel with copper or mixed metal bolts at the same interval apart, and from $\frac{7}{8}$ inch to $1\frac{1}{2}$ inch diameter (see Tables from Lloyd's Rules). In vessels built on a temporary keel, the fastening of the keelson must be delayed until the permanent keel is laid.

The other pieces of the framing of the ship are set up in the order that may be most convenient; observing only, in wooden ships, that as some of those pieces have to be fastened through the outside planking, their permanent fastening must be delayed until after the outside planking has been put on.

Wooden ships of the Royal Navy are often left "in frame" to season from six months to a year, before the planking is put on.

In raising the pieces of the framework of a ship to their places, sheer-legs are commonly used in an open slip; and in a shed, a travelling crane is sometimes employed.

59. *Putting on the Skin*.—Preparatory to putting on the skin, the *sheer lines* of the sides, and *normal* lines of the bottom, are to be marked on the outside surface of the frames, in order to regulate the position of the seams of the plating or planking. These lines may be laid off from the *expansion of the skin* (Second Division, Articles 26, 27, 31); but the normal lines may also be easily and accurately constructed on the framing of the ship itself, by pinning a broad, straight-edged, and flexible batten to the frames, so as to cross the midship frame at right angles; when it will of itself assume the true figure of a normal line. Such lines are required to be laid off on wooden ships at every sixth or eighth seam.

In *iron ships* the first operation in putting on the skin is to fit the inner strakes of plating (being those which lay to the frames) in their proper places, fastening them in a temporary way with bolts or pins. They are then taken down, and the rivet-holes are punched or drilled through them; then set up again, and rivetted to the frames; and the butt-straps are also rivetted to the inside of them at their butt-joints. The outer strakes are next fitted on, punched or drilled, and rivetted to the frames through the filling pieces (which should be strips of plate completely filling the spaces between the frames and the outer strakes), to the inner strakes at the seams, and to butt-straps inside the butts (see Articles 49 and 50 of this Division). The operation of rivetting will be described further on.

The outside planking of a *wooden ship* cannot be permanently fastened until the inside planking is brought on also, because the fastenings have to pass through both inner and outer skin. It is therefore first put on with temporary bolts or screws; when each strake has been accurately fitted to its place, the treenail-holes and bolt-holes are bored through the planking and frames: then the inside plank is brought on, the holes bored through it, and the fastenings driven. The planking of a wooden ship is sometimes left for a time with temporary fastenings only after the holes have been bored, in order to season it the better; and during that time a few strakes are left out, to make openings for the circulation of air.

Planking is said to be *single-fastened*, *double-fastened*, or *single and double fastened*, according as each strake is fastened to each frame that it crosses at one point, two points, or one and two points alternately (see Plate $\frac{H}{2}$, from Lloyd's Rules for Wooden Ships). There is reason to believe that single-fastening is really sufficient in all cases, when the frame is sufficiently braced diagonally. The outside planking between wind and water, and

that of the top-sides, is coated to the frames, to enable it the better to resist tension.

Seams of iron plating are sometimes single-rivetted, and sometimes double-rivetted: butts are almost always double-rivetted, and sometimes treble or quadruple rivetted. It may be doubted, however, whether anything more is really necessary than single-rivetting for the seams, and double-rivetting for the butts.

The usual lap of plates is from five to six diameters of the rivets at double-rivetted joints, and about three diameters at single-rivetted joints; the pitch of the rivets, four diameters in seams and butts of plates, and eight diameters in angle-irons and other bars.*

60. *Rivetting*.—(See Division Third, Article 60, and Article 50 of the present Division.) Some of the rivetting for iron ships (such as the fastening of flanges to the webs of beams, &c.) may be performed by the rivetting machine, because the work can be carried to the machine; but by far the greater part of it is done by hand.

The rivets are heated in a small portable furnace, called a *rivet-hearth*, which can be carried to any part of the ship. The burning fuel is contained in a shallow round iron tray, supported by three slender legs: below the tray is a pair of small circular smiths' bellows, to blow the fire. A set of rivetters consists of two rivetters, to clench the rivets on the outside of the plating; a holder-up, to put the rivets through the holes from the inside, and hold them steady against the blows of the rivetters' hammers; and a boy, or sometimes two, to blow the fire, and hand the rivets to the holder-up. Mr. Grantham estimates the number of rivets driven by a set of rivetters, in a day's work of ten hours, at about 100, if employed by the day, or 140, if employed by piece-work. The rivets should be at a bright-red heat during the whole process of rivetting-up, which should be done very rapidly, that they may not have time to cool.

The clenched end of the rivet when finished should be flush with the outside plating; and should any part project, through being more than sufficient to fill the countersunk hole, it is to be cut off with a chisel.

61. *Fastening by Bolts*.—As already stated in Article 61 of the Third Division, all bolts that are used as fastenings for timber pass through washers or "*rings*," at their ends, usually of the same metal with the bolts, to give them a proper hold of the timber.

A bolt which is to be capable of being unfastened may be secured either by means of a screw and nut, or by means of a pin or wedge called a "*forelock*," passing through a hole in the end of the bolt. The screw and nut, if properly proportioned, may be made nearly, though not quite, as strong as the body of the bolt; the forelock is not more than half the strength.

Bolts for permanent fastenings are secured, after having been driven, by being clenched upon a ring with the hammer.

Copper and yellow metal are the only suitable metals for the fastening of outside planking. Where such bolts are used throughout instead of treenails, Lloyd's Rules prescribe that at least two-thirds of them must be driven through and clenched: the remaining third may be short bolts, called *dump bolts*, stopping in the frame timbers: their lengths have been given in the Appendix

to the Third Division. The same rules prescribe that every butt in the outside planking shall be fastened with two bolts, one of which must be through and clenched; and that bolts through and clenched are alone to be used as fastenings where important pieces of the framing are connected together.

The heads and rings of the bolts are sunk into the outside planking.

62. *Fastening by Treenails*.—Treenails, after having been driven through, are in a manner clenched at the ends by *caulking*: that is, two, three, or four cuts, according to the size of the treenail, in the form of a cross, triangle, or square, are made in its ends with a chisel, and are then caulked with oakum, so as to spread the ends.

According to Lloyd's Rules, two-thirds of the treenails are to pass quite through both skins; the other third may be short, like dump bolts.

Screw treenails, having a screw thread cut in them, have been used by Messrs. Hall of Aberdeen.

63. *Fastening of Decks*.—Plate-iron decks, or parts of decks, such as water-ways, stringers, and diagonals, are rivetted to iron beams.

The deck planking, or flat of the deck, is bolted to iron beams with galvanized iron bolts, whose usual dimensions have been exemplified in the Appendix to the Third Division. When the beams are of timber, the planks are usually fastened to them with *deck-nails*, two to each strake and beam; which nails, for the *weather-decks*, or exposed decks, are of mixed metal, and for other decks, of iron. (As to the dimensions and weight of nails in general, see the Third Division, Article 62.) Treenails are sometimes used for the same purpose.

The construction of decks for carrying heavy guns will be further considered in the Seventh Division.

64. The *Filling* of spaces between frames, in iron ships with cement, brick-work, and other materials, and in wooden ships with pieces of wood, has been mentioned in Article 46 of this Division.

In iron ships, this operation is performed after the outer skin has been put on. If cement is used, it should be carefully kept dry until wanted: its quality should be ascertained by mixing it with the proper proportion of sand (about half the bulk of the cement), and observing whether a specimen *sets*, or hardens rapidly under water. The sand should be clean and sharp. The sand and cement should be thoroughly well mixed immediately before being used. Care should be taken that too much sand is not added, as it will make the mixture brittle; that the mixture is used at once, and not left standing; that every part of each space is thoroughly filled with it; and that it forms a complete coating to the iron of not less than half an inch in thickness at the thinnest parts.

When bricks are used, they should be chosen for hardness, regular figure, smooth surfaces, and straight sharp edges, a clear ringing sound when struck together, and compactness and uniformity of material, as shown when broken. They should be built in cement, mixed as already described; and each brick should be soaked in water before being laid, otherwise it will dry the cement too fast and make it crumble. Gas coke is used in some cases instead of bricks, for lightness in large spaces.

When asphaltic mastic is used for filling, it may be made either by combining seven or eight parts of powdered natural asphalt in a boiler with one part of bitumen, at a heat sufficient to liquefy

* With a view to the preservation of the plates, it is a good practice, before putting them on, to coat them either with a drying oil by the method described in Article 8 of this Division, Process II., or with zinc (galvanizing).

the asphalt, or by making an artificial asphalt of coal tar, mixed with finely ground limestone or fireclay, until a composition is obtained which, when cool, is just soft enough to yield visibly to the pressure of the nail. The asphaltic mastic may be mixed with about one-half or three-fifths of its volume of sand, and then used for filling spaces, either alone or along with bricks, which, before being built, should be heated in an oven, and dipped in coal tar.

Fillings of wood between wooden frames are put in before the outside planking is brought on. They are made of any small pieces of sound timber which may be available, and are laid with the grain ranging the same way with that of the frames: they are carefully fitted to their places, and the joints where they touch the frames and each other are caulked.

65. *Caulking* the seams and butt-joints of an *iron ship* is performed by means of tools like blunt chisels: these are of two kinds—a blunter tool for the butt joints, and a less blunt tool for the seams. With a hammer and one of those tools, the caulker makes an indented groove parallel to, and about an eighth of an inch from the joint to be caulked: at the seams, that groove is on the edge of the overlapping plate; at the butt joints, on the outer surface of the plate, on each side of the line in which the plates meet. The effect of this process is to force the particles of iron towards the joint, and thus to close it tightly. Water-tight iron bulkheads and platforms are caulked like the outside plating.

In a *wooden ship* caulking is performed, first on the treenails, and then on the planking, by driving into the seams *threads* or layers of *oakum*. Oakum is made by cutting old ropes and cables into short lengths, called *junk*, which are then picked to pieces.

The seams of the planking, in order to receive the oakum, require to be open to the extent of about $\frac{1}{10}$ of the thickness of the plank. A gauge for the proper width of opening is made by setting the arms of a jointed rule to the angle produced by $\frac{1}{2}$ inch of opening between them at a point 10 inches from the joint; when the opening between the arms of the rule at a distance from the joint equal to the thickness of plank will show the proper width of opening for the seams at their outer edges.

Seams that are closer than the proper width are then *opened* to it, or *reamed*, by driving into them sharp iron wedges, called *reaming irons*, with a mallet, called a *beetle*. This is considered a test of the sufficiency of the fastenings; and should they prove insufficient, so that planks are started by the opening, new fastenings are at once to be put in. After each seam has been opened, the proper number of threads of oakum are forced in one after another with an iron wedge, called a *caulking iron*, driven by a beetle or mallet; beginning usually with spun yarn or with white oakum, and finishing with black oakum.

The following table shows the number of threads of oakum used in caulking the seams of planking of different thicknesses:—

WALES AND BOTTOM PLANK.

Thickness—Inches,.....	1	2	2½	3	4	5	6	7	8	9	10
Oakum—Double Threads,.....	1	2	3	4	5	6	8	10	11	12	13
Spun Yarn—Single Threads,....	—	—	1	2	2	2	2	2	2	2	2

TOPSIDES AND WATER-WAYS.

Thickness—Inches,.....	2½	3	4	5	6	7	8	9
Black Oakum—Double Threads,...	2	3	4	5	7	9	10	11
White Oakum—Double Threads,...	1	1	1	1	—	—	—	—

GUN-DECKS.

Thickness—Inches,.....	3	4
Black Oakum—Double Threads,.....	2	3
White Oakum—Double Threads,.....	1	1

WEATHER-DECKS.

Thickness—Inches,.....	2	2½	3
Black Oakum—Single Threads,.....	1	2	2
White Oakum—Single Threads,.....	1	1	1

The number of threads of oakum for caulking the side and bottom may be calculated by rule, as follows:—*Take the nearest whole number to once and a quarter the thickness of plank, in inches.*

After the oakum has been driven in, it is further compressed by means of a tool called a *making iron*, or *horse-iron*, held by one caulker, and struck with the beetle by another: this is called *horsing-up*. It is then *payed* with melted pitch; and then the seam is filled up flush by laying in a thread of spun-yarn.

Weather-decks (that is, decks not covered) are payed with *marine glue*—a solution of caoutchouc and shellac in naphtha.

The opening and caulking of one seam tends to close the seams near it; so that although some of them may have previously been above the proper width, they may still require to be opened before caulking.

The opening and caulking of seams requires careful superintendence, to see that the seams are really opened to the bottom, and not merely notched into shallow grooves; and that the oakum is really driven home to the bottom of the seam, and not *choked*, or wedged into a mass near the surface, leaving the bottom empty.

Butts, and the vertical scarfs of the keel, are caulked like seams; and so also are any rents or shakes which may occur in the planking.

In caulking fir decks, the operations of reaming and horsing-up are generally omitted, because of the softness of the wood.

66. *Protection of Wooden and Composite Ships*.—The bottoms of wooden and composite vessels, up to the water-line, and sometimes a little further, are usually protected by copper or mixed metal *sheathing*, and their sides and upper works by paint. The bottoms of ships, before being sheathed with metal, are *payed* over with pitch or tar. Small vessels for coasting trade are sometimes payed with pitch or tar only, and not sheathed.

The chemical properties of copper and mixed metal used for sheathing, have been stated in Articles 10 and 11 of this Division.

The metal for sheathing is usually made in sheets of the following weights and dimensions:—

Weight in ounces per square foot,.....	18	28	32
Thickness in inches, about.....	0·025	0·038	0·044
Length, inches,.....	48	48	48
Breadth, inches,.....	20	14	14

Such sheets are named according to the number of ounces to the square foot. Thirty-two ounce sheathing is used for the bows, and for the parts between wind and water; twenty-eight ounce sheathing for the rest of the bottom; and eighteen-ounce sheathing for the the lower side of the main keel, between it and the false keel. The sheets are put on in strakes, running fore and aft; the after end of one sheet overlapping the forward end of the next, and the lower edge of one strake overlapping the upper edge of the next below; and they are fastened on with mixed metal nails, called *sheathing nails*.

The rate at which the corrosion of copper sheathing goes on, is extremely variable in different cases, ranging from less than an ounce to ten or twelve ounces per square foot in a year. (See foot-note to Article 67.) The smallest perceptible rate of corrosion is sufficient, by the scaling off of the oxide, to keep the

ship's bottom free from barnacles and weeds; but it is not advisable to protect the sheathing completely against corrosion (as was done by Sir Humphry Davy's protectors, consisting of bands of an electro-positive metal, such as iron or zinc); for then the bottom rapidly becomes foul, to the injury of the vessel's speed.*

The sides above the sheathing, and the other woodwork of a ship, usually receive three or four coats of paint. All wood should be thoroughly dry when painted.

67. *Protection of Iron Ships.*—The process of galvanizing, or coating with zinc, forms a very efficient protection for iron against oxidation in sea-water, as well as in air. The processes I., II., and III., mentioned in Article 8 of this Division, by which iron plates are coated with pitch, drying oil, and paint, have also a considerable effect in protecting the plating of an iron ship against oxidation, especially if the iron is coated with oil while hot, and afterwards painted. The paint used should be such as will not of itself tend to oxidate the iron; red lead paint is therefore objectionable, for it contains a large proportion of oxygen combined with lead; and as lead is electro-positive to iron, the oxygen tends to quit the lead and combine with the iron. Zinc paint has no such effect; for zinc is electro-negative to iron. The ironwork of ships usually receives three coats of paint outside, and two inside.†

Care should be taken to avoid the use of paint thickened with the white powder of sulphate of barytes (heavy spar) as a cheap imitation of oxide of zinc: that substance is without chemical action on the iron; but it injures the tenacity and compactness of the paint, and makes it pervious to water and air, and liable to crumble off after drying.

Merely protection against oxidation has no effect in preventing iron ships from growing foul, by the adhesion of shells, and afterwards of sea-weed; for that purpose a coating is required which shall peel off by slow degrees, carrying the barnacles away with it, and shall neither be too durable, which would enable the barnacles to adhere, nor too perishable, which would cause it to

* Experiments have been made by Major-General Sir Arthur T. Cotton on the comparative resistance of paint and of metal sheathing on the bottoms of vessels. In the course of those experiments it was found, that sheathing the bow of a vessel with copper produced a diminution of the friction to little more than one-half of that on a painted surface, while sheathing the run with copper produced no sensible diminution whatsoever.

This appears to prove, that the run of a vessel has a skin or shell of water adhering to it and following it; so that while at the entrance, the friction to be overcome is that of water gliding over a surface of paint or of metal, as the case may be, the friction to be overcome at the run is that of one layer of water gliding past another.

Hence the "coefficients of propulsion" deduced from experiment in the First Division of this Treatise, Articles 164 to 167, such as 20,000 for clean painted iron ships, and 21,800 for a coppered wooden ship, probably correspond to coefficients of friction intermediate, in the one case, between the friction of water on water, and that of water on paint; and in the other, between the friction of water on water, and that of water on copper.

† Dr. Crace Calvert and Mr. R. Johnson have made experiments on the loss of weight undergone by plates of various metals and alloys when exposed for one month to the action of sea-water. The details will be found in the "Transactions of the Literary and Philosophical Society of Manchester for 1865." The following Table gives an abridgment of the general results, reduced to fractions of a pound (avoirdupois) per square foot of immersed surface per month:—

	In a vessel of Sea-water.	In the Sea.
Steel,.....	0060	0216
Iron,.....	0056	0204
Copper (best selected),.....	0027	0061
Do. (rough cake),.....	0028	
Zinc,.....	0012	0070
Galvanized Iron,.....	00028	0030
Tin,.....	0003	
Lead,.....	trace	0058
Brass (Copper 50, Zinc 50),.....	0024	
Do. (Copper 50, Zinc 48, Tin 2),.....	0022	
Do. (Copper 66, Zinc 32.5, Iron and lead 1.5),.....	00155	
Muntz's Metal—Sheet (Copper 70, Zinc 29.2, Iron and Lead 0.8),.....	0015	
Muntz's Metal—Bars (Copper 62, Zinc 37, Lead and Iron 1),.....	0014	

The loss of weight of lead exposed to the sea is ascribed chiefly to mechanical wearing. In brass, the presence of tin appeared to protect the zinc and increase the corrosion of the copper; the presence of iron seemed to protect both copper and zinc.

be too soon worn out, and lead to too great expense for its renewal. Copper and yellow metal have exactly those properties; but they cannot be directly applied to an iron ship, because of their being electro-positive to iron, and causing it to corrode rapidly. Various compositions have been used with more or less success, such as lime soap, amalgam of mercury and zinc, paint containing metallic copper in powder, or red oxide of copper, insulated from the iron by the oily matter of the paint, &c.

To enable iron ships to be sheathed with copper or yellow metal, it is necessary that the sheathing should be insulated from the iron. This has been effected by Mr. Grantham in the following manner: outside the iron skin are rivetted angle-iron ribs, whose projecting flanges are of a dovetail shape in section. An equal weight of iron is saved in the inside framing. The iron skin is then coated with pitch, and the spaces between the dovetail flanges are filled by packing and wedging into them short pieces of plank. The outside ribs with their wooden filling rise to a short distance above the water-line, and the upper edge of the filling is guarded by a longitudinal angle-iron. The outer surface of the fillings having been payed with pitch, a complete wooden sheathing, about 1½ inch thick, is put on, and fastened to the filling pieces with mixed metal nails, which should not pass through those pieces. The wooden sheathing is then pitched, and is sheathed with copper or mixed metal in the usual way; care being taken to keep the metal sheathing two or three inches from any exposed piece of iron.

Mr. Daft's method of sheathing iron ships with copper, mixed metal, or zinc, is as follows:—The inner layer of the iron skin consists of narrow strips of plate, merely wide enough to make lap joints with the outer layer, and to leave a groove between the edges of each pair of outer plates, about as wide as the plates are thick. Into that groove is inserted a filling of teak or of ebonite (a hard compound of caoutchouc and sulphur). Outside the plating is a layer of tarred felt, about ½ inch thick, upon which the sheathing is laid, and fastened with sheathing nails of the same metal, driven through the felt into the teak or ebonite fillings. Intermediate fastenings are obtained, if required, by inserting ebonite plugs into holes drilled in the iron plates, and driving sheathing nails into them through the felt.

The tarred felt serves to insulate the copper or mixed metal from the iron. It may be used with zinc sheathing also, but is not then absolutely necessary; for zinc, being electro-positive to iron, protects the iron against oxidation.

During some experiments made in 1864 at Shoeburyness, it was found that *zinc sheathing upon iron* lost about .002 inch of its thickness by six months' exposure to sea-water, and remained free from shell-fish and sea-weed, like copper or yellow metal.

68. *Boat-building.*—Boats are almost always built of wood. The best kinds of timber for them are the same with those which are suited for the parts of ships that are alternately wet and dry. According to the manner of building them, they are distinguished into *Carvel-built*, *Clinker-built*, and *Diagonal-built* boats.

In all three of those styles of boat-building, there are a keel, stem, and stern-post, rabbeted to receive the planking, as in a ship; the stem is scarfed, and the stern-post tenoned, to the keel.

I. *Carvel-built* boats are built like ships in miniature. They have frames, each generally consisting of a floor and two futtocks; the floors are scored down over the keel, and fastened to it with bolts in the larger and nails in the smaller boats. The frames are

sided, moulded, and trimmed to their proper bevellings, like those of a ship, and are kept temporarily in their shapes and places by cross-spalls, ribands, harpins, and shores. The planking consists of strakes laid fore and aft with flush seams, like those of a ship; they are usually fastened with two nails in each timber of the frame. The strakes first put on are the lowest, or *garboard strake*, and the uppermost but two, called the *binding strake*. Above the binding strake is the *landing strake*; the *gunwale* rests on the timber heads, and covers the upper edge of the landing strake; and the uppermost, or *sheer-strake*, has its upper edge flush with the top of the gunwale, and its lower edge overlapping the landing strake. The stern is usually strengthened by a *transom*, and the bow by two *hooks*. Moveable strakes above the gunwale are called *wash-strakes*.

The *thwarts* are the transverse planks which keep the sides asunder, like the beams of a ship, and serve as seats for the rowers; some are fixed, and others loose; the fixed thwarts are secured to the sides with knees. The thwarts are spaced about 2 feet 10 inches, from centre to centre, in single-banked boats, and 3 feet in double-banked boats.

Some boats have a fixed inside planking, or *footwaling*, in the bottom; others have moveable *bottom boards*; others, *gratings*.

II. *Clinker-built* boats are the lightest class for their strength and size; they are distinguished by the lower edge of each strake of plank overlapping the upper edge of the next strake below. They are not built upon frames, but upon temporary transverse sectional moulds, two, three, or four in number, which are fixed at their proper stations on the keel; the strakes are then put on,

beginning with the garboard strake, and bent to the figure given by the moulds: each strake is fastened to the next below it by nails driven from the outside through the *lands* or overlaps. When two or more lengths of plank occur in a strake, they are scarfed to each other, the outside lip of each scarf pointing aft. The scarfs have a layer of tarred paper between, and are fastened with nails driven from the thin end of each piece. Towards the hooding-ends, the strakes are *chased* into each other; that is to say, a gradually deepening rabbet is taken out of each edge at the lands, so that the projection of each strake beyond the next below it gradually diminishes, and they all fit flush with each other into the rabbets of the stem and stern-post. Floors, futtocks, and hooks, are afterwards put in, and fastened to the planking by nails driven from the outside, and clenched inside.

III. In *Diagonal-built* boats the skin consists of two layers of planking, with flush seams, making angles of about 45° with the keel, in opposite directions. They are built, like clinker-built boats, upon temporary transverse moulds. After setting up and fixing the moulds upon the keel, the gunwale, a shelf-piece, and a series of ribands are temporarily fixed on the moulds. The two layers of planking are then put on, bent to fit the moulds and ribands, and fastened to each other and to the keel, stem, stern-post, shelf, and gunwale with nails, driven from the outside, and clenched inside upon small rings. The gunwale is then shored to keep it in shape; the moulds and ribands are taken out, and floors, hooks, thwarts, &c., are put in, as in a clinker-built boat.

As boats precisely similar in all their parts^a are made in large numbers, machinery has been applied to their manufacture.

CHAPTER V.

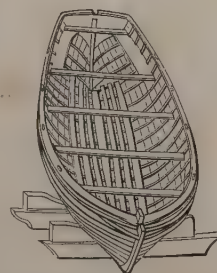
VARIOUS EQUIPMENTS OF SHIPS.

69. *Rudder*.—The form and dimensions of the rudder, as affecting its action on the water, have already been considered in the First Division, Articles 187, 188; its strength has been treated of in the Third Division, Article 86; and the rules commonly followed as to the scantlings of its principal parts, have been given in the Appendix to that Division. The pieces of which the rudder consists, and the way in which they are put together, remain to be described.

An *iron* or *steel* rudder usually consists of a frame, covered on both sides with flush-jointed plating; the two layers of plating being rivetted through the frame to each other with rivets countersunk at both ends, and also rivetted at their seams to covering straps inside. For illustrations of the framing in an iron ship see Plate $\frac{1}{2}$; and in a steel ship, Plate $\frac{1}{4}$. The foremost piece of the framing is the *rudder-stock*; its upper end, called the *rudder-head*, is cylindrical, and rises through the cylindrical *rudder-port*, and through a vertical tube or rudder case, having a stuffing box at the top, into the stern of the vessel; its lower end, or *heel*, usually forms a pivot turning in a hole in the *skeg*, or projecting after-end of the keel; its intermediate part is square, and (except in the balanced rudder) is hinged by pins, called

pintles, fitting into eyes, called *braces*, to the stern-post, or to the rudder-post. The aftermost piece of the framing is curved to the shape of the after-edge of the rudder, and is usually welded at its upper and lower end to the rudder-stock. The cross-pieces are in general opposite the pintles, and are welded to the rudder-stock and the after-piece of the framing. In fitting iron rudders it is usual for one or two of the eyes or braces to have the holes for the pintles drilled only partially through them. Into each hole is fitted a steel pin, with the upper surface spherical. The corresponding pintle is fitted with a steel pin having its lower surface spherical. When

Fig. 2 A.



^a According to a method of constructing boats introduced by Mr. George Fawcus, the outer and inner surfaces of each boat are so shaped, that any number of boats of similar figure and equal size can be packed one inside another in any order, the projecting lower edge or land of the sheer-strake of one boat resting on the gunwale of the boat next below. The thwarts are all moveable, and when in their places in a boat, are fastened by means of oval pins to fixed iron knees. When a set of boats are packed together, the rudders of all the boats, and the thwarts of all except the uppermost, are unshipped, and laid in the uppermost boat. Fig. 2 A gives a bird's-eye view from ahead, of two boats, packed one inside the other.

the rudder is shipped, its weight is borne at the points of contact of these spherical surfaces, so that the friction is very small.

For an illustration of the manner in which a *wooden rudder* is hinged with pintles and braces of mixed metal to the stern-post, see Plate $\frac{6}{5}$. The rudder consists of an assemblage of

pieces of timber, coaked and bolted together like those of the framing of the stem and stern-post. Fig. 1 shows the usual arrangement of those pieces. The dotted line, XX , represents the axis of motion of the rudder, being the common axis of all the pintles and braces, and of the rudder-head. MM is the *main-piece*, usually of oak or other timber of equal quality. Its upper end is cylindrical, and is the *rudder-head*. The shoulder, C , where it is in contact with the head of the stern-post, is conical; and at the small end of the cone is the uppermost of the pintles, which are all marked P . The other pintles are fitted into scores in the foremost piece of the rudder, F , which is usually of elm. The form of the after-part of the rudder is made up with the pieces A, A , usually of fir. At the bottom of the rudder is the *sole-piece*, S , usually of elm, lightly fastened on, so that, like the false keel, it may be knocked off without further injury to the rudder.

Fig. 2 is a horizontal section of the foremost edge of the rudder, R , and the after-most edge of the stern-post, SP , showing how they are bevelled or *bearded*, so as to admit of the helm being put

over either way to the usual greatest angle of 42° ; and how the shoulders of the pintles, and the wood above and below them, having cylindrical surfaces described about the axis of motion, X , fit into a cylindrical hollow in the stern-post.

Wooden rudders are *sheathed*, like the ship's bottom, with copper or yellow metal.

Before the rudder is hung, the braces on the stern-post are adjusted to their correct positions by passing through them a perfectly straight cylindrical wooden rod, of the same diameter with the pintles.

The rudder when hung is guarded against being *unshipped* (or thrown upwards out of its place), by a moveable piece called a *wood-lock*, which is screwed upon the stern-post or rudder, and fits into a score a little below the uppermost pintel.

In the First Division, Article 188, a general description has been given of Mr. Lumley's rudder, consisting of a *body* hung to the stern-post in the usual way, and a *tail* hung in the same manner to the body, and moved in such a way, that when the body is put over to a given angle, the tail is at the same time put over to about double that angle.

There are different ways of effecting this by mechanism; the simplest is that illustrated by the skeleton plan, Fig. 3, in which X is the axis of motion; XR the body, and RT the tail. To the tail is fixed a *yoke* or arm, RY , connected by a link, YB , with a fixed projecting bracket, which is fastened to the stern of the vessel; the effect when the helm is put over is shown by the dotted lines.

The breadth of the body is from one-half to three-fourths, and the breadth of the tail from one-half to one-fourth, of the whole breadth of the rudder.

The following is the construction for adjusting this apparatus

so as to work in the most correct manner. XRT being the mid-ship position of the rudder and tail, bisect XR in L . Lay off the equal angles $LXB = LRY =$ one-half of the greatest angle to which the *tail* is to be put over; make $XB = RY = XL$; and draw the straight line BLY . Then RY will represent the yoke; YB the link; and B the position of the point where the link is to be jointed to the bracket.

The link may pass either over the top of the body of the rudder, or through a hole in it.

This rudder may be converted, when required, into a common rudder, by unshipping the link, and dropping over the tail a strap or bridle, which fixes it to the body.

The rudder-head in every case turns in a collar in the uppermost of the decks which it traverses. In small vessels this is often the weather-deck; in large ships, and especially in ships of war, it is usually the gun-deck, or lowest deck, that is permanently above water.

Rudder-chains, and *rudder-pendants*, are chains or ropes which are shackled to a bolt at the after edge of the rudder, immediately above water, and fastened to bolts at the ship's quarter. They hang slack enough to permit the free motion of the rudder. Their use is to prevent the rudder from being lost, in the event of its being unshipped; and sometimes also they are led in-board, and used for steering, in the event of the tiller or rudder-head giving way.

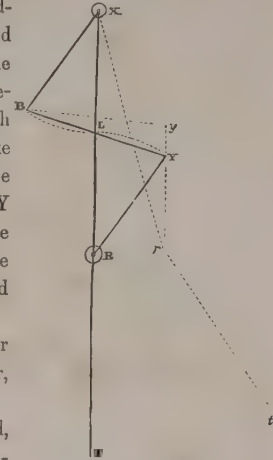
70. The *Helm* comprises the whole steering apparatus. Besides the rudder, it most frequently consists of a *tiller* or a *yoke* (as the case may be), a *steering-wheel*, and ropes or chains (or, in some cases, screws and nuts) to transmit motion from the wheel to the tiller or the yoke. Small vessels and boats only are steered by the tiller alone, or by a yoke with ropes held in the hand.

Wheel-ropes are often made from strips of untanned hide, kept dry and well greased; such being stronger than hempen ropes.

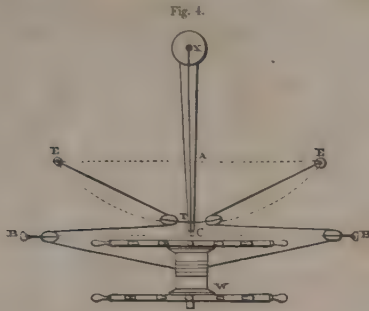
The tiller is a lever fixed to the rudder-head, and pointing in most cases forward, and in some cases aft. (For examples of tillers pointing aft, see Plates $\frac{4}{5}$, $\frac{7}{5}$, and $\frac{8}{5}$.) The yoke consists of a pair of arms pointing sideways in opposite directions. The principles upon which the strength of the tiller or of the yoke depends, have been stated in the Third Division, Article 86.

When rope or chain tackles are used for putting over the tiller, their ordinary arrangement is that shown in Fig. 4; in which X is the rudder-head, and XT the tiller, having a pair of single blocks fixed to its forward end, T . E, E are a pair of eye-bolts in the deck, to which are made fast the ends or *standing parts* of the steering-chains or wheel-ropes. Those chains or ropes are led through the blocks of the tiller, already

Fig. 3.



mentioned; then through a pair of fixed blocks, B, B, attached to the deck; then through another pair of fixed blocks beneath, or nearly beneath, the barrel of the steering-wheel, W, and



hidden by it in the figure; then through holes or tubes in the deck or decks that lie between the tiller and the wheel; and then round the barrel, to which they are made fast at the middle of its upper side. The total number of turns of the steering-chains round the barrel is usually *five, seven, or nine*, so that from *two and a half to four and a half* turns of the steering-wheel are required to put the helm hard over to starboard or to port. The length of chain wound on the barrel on turning it either way with the single purchase, is about double the length of the arc through which the end of the tiller is put over; and the effective diameter of the barrel (being = its actual diameter + the diameter of the chain) is adjusted accordingly. Sometimes there is but one steering-wheel, and sometimes there are two, at opposite ends of the barrel. Steering-wheels range from 3 to 6 feet in diameter, and are made of mahogany, or timber of similar quality, strongly framed together, and bound with brass.

In every case the rudder is to be so connected with the steering-wheel, that in putting the helm over, the *lower* rim of the wheel shall be moved in the opposite direction to the rudder: that is, in the same direction with a tiller pointing forward.

Hence, when the tiller points *forward*, the steering-chains pass *over* the barrel first; and when it points *aft*, *under* the barrel. The arrangement of tackles, when the tiller points aft, is illustrated in the upper figure of Plate 2.

When it is desired to have a steering-wheel on the bridge-deck, or any other part of the ship that is distant from the rudder, the steering-chains may be led from the blocks, B, B, to the barrel of that wheel through tubes and round sheaves arranged in any way which may be convenient; but to provide for the chance of such apparatus getting out of order, it is always advisable to have also a steering-wheel in the usual position, near the rudder.

Fig. 4 of the present Chapter shows the geometrical construction (as described by Mr. Peake, in his "Treatise on Shipbuilding,") for finding the positions where the eye-bolts, E, E, and blocks, B, B, are to be fixed, in order that the slackening of the chains when the helm is put over may be the least possible. About X, the axis of the rudder, with the radius, \overline{XT} , the effective length of the tiller, describe a circular arc. Take $\overline{TA} = \frac{1}{3} \overline{XT}$, and at A draw a straight line perpendicular to \overline{XT} , cutting the circular arc in E, E; these points will be the stations for the eye-bolts. Then produce \overline{XT} to C, making $\overline{TC} = \frac{1}{3} \overline{XT}$; and through C draw \overline{BCB} perpendicular to

\overline{XC} , making $\overline{CB} = 1\frac{1}{2} \times \overline{AE}$. Then B, B will be the points to which the blocks are to be made fast.

The method usually adopted in H.M. service for getting rid of slack rope, is to make the diameter of the barrel smaller at the middle than at the ends, so that in moving the rudder from amidships to the extreme position, the excess of rope wound on the barrel over that unwound is equal to the rope which would have been slack had the form of the barrel been cylindrical.

The arms of a yoke are usually connected by rope or chain tackles with a pair of ring-bolts at the stern, so that by hauling on one of those tackles, the corresponding arm of the yoke is pulled aft; from the fixed blocks of those tackles the ropes or chains are usually led straight ahead; then under a pair of sheaves in blocks fixed to the deck; then up through vertical tubes to the barrel, *over* which they first pass, for the reason formerly stated as to the direction of motion of the wheel. The use of a yoke becomes necessary in vessels which have a screw capable of being unshipped and lifted into a vertical trunk in the stern; because that trunk occupies the place where the tiller would move.

In merchant ships the steering-wheel, instead of a barrel, has sometimes on its axis a right and left handed screw, with two nuts, which are acted upon by the right and left handed threads respectively, and are connected by means of suitable links with the two arms of a yoke—the left-handed nut with the starboard arm, and the right-handed nut with the port arm. When the lower rim of the wheel is turned to starboard, the left-handed nut, with the starboard arm of the yoke, is driven aft, and the right-handed nut, with the port arm of the yoke, pulled forward. When the lower rim of the wheel is turned to port, the right-handed nut, with the port arm of the yoke, is driven aft; and the left-handed nut, with the starboard arm of the yoke, pulled forward. The links for connecting the two arms of the yoke with the nuts should be of exactly equal length, and if oblique, of exactly equal obliquity; otherwise the apparatus will work incorrectly, and be liable to jam.

An apparatus for *steering by steam-power*, invented by Mr. Sickels, has been in use since 1860, it is said, with good results. The steering-wheel, and the barrel for the steering-chains, are upon separate shafts, in a line with each other. On the shaft of the steering-barrel is a toothed wheel, gearing with a pinion upon a crank-shaft, which is driven by a small steam-engine having two cylinders working at right angles to each other. The eccentrics which work the slide-valves of that engine are upon a separate shaft, having a pinion similar to the former pinion, driven by a toothed wheel similar to the former wheel, which latter toothed wheel is fixed to, and moved by, the steering-wheel. Thus, when the steering-wheel is turned, it causes the eccentric shaft to turn, and every revolution of the eccentric shaft causes the engine to make one revolution; and thus the motions of the barrel are made to correspond exactly with those of the steering-wheel. Another steering-wheel is made fast to the barrel, to be used in the common way, in the event of the engine getting out of order; in which case the wheels and pinions are thrown out of gearing.

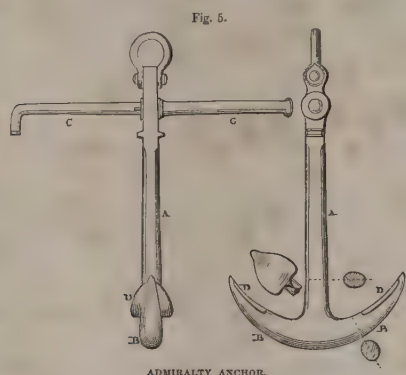
71. *Anchors*.—The subject of the strength of anchors has been considered in the Third Division, Article 87; and their weights and proof-loads for ships of different sizes, according to the ordinary rules, have been given in a table in the Appendix

to that Division. The full complement of anchors for a large ship consists of six, and sometimes seven: two, called the *bower anchors*, for ordinary use in a roadstead; two, called the *sheet anchors*, of the same size with the bower anchors, kept in reserve in case the bower anchors should be lost; one smaller anchor, called the *stream anchor*, for riding in sheltered places; one smaller still, called the *kedg anchor*, or *kedg*, used for warping the ship along a river channel; and sometimes a second and smaller kedg anchor.

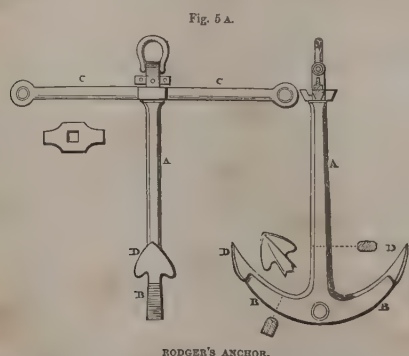
The usual shape and arrangement of anchors are illustrated in the upper-deck plan, $\frac{B}{5}$, and longitudinal section, $\frac{B}{7}$, of H.M.S. *Warrior*; and in the longitudinal section, $\frac{C}{7}$, of H.M.S. *Victoria* and *Albert*.

Figs. 5 and 5 A show the principal parts of which an ordinary anchor consists.

A is the *shank*, having at the smaller end the *ring* or *shackle*, which is fastened to the shank with a bolt passing through a round eye, and secured by a forelock. The length over all,



including the shackle, ranges from about 8 feet to 18 feet; and the weight of the anchor, in cwts., (exclusive of the stock) may be roughly estimated at about *one-fiftieth part of the cube of the length in feet*. To deduce the length from the weight in cwts., *multiply by 50 and extract the cube root of the quotient*.

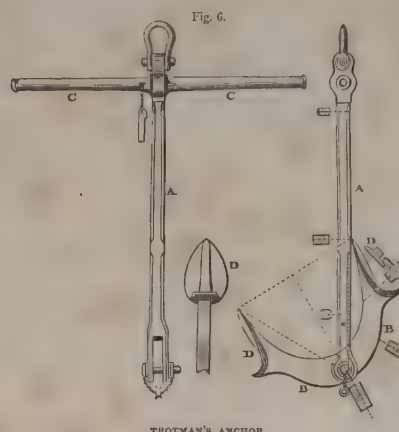


B, B are the *arms* for taking hold of the ground, forged in one piece with the shank, and terminating in the *flukes* or *palms*, D, D. The inner surface of each of the arms is shaped nearly like a circular arc of from 50° to 70° , with a radius equal to about half the length of the shank, or rather less.

Fig. 6 represents a kind of anchor called a *hinged anchor*, in which the piece forming the two arms is separate from the shank, to which it is bolted through a round hole, and each of

the flukes has a horn or projection at the back to make it take hold of the ground the more surely.

The *crown* is the place where the two arms unite; the *throat* is the adjoining part of the shank; the *trend* is that part of the shank which extends from the throat to a distance equal to the



length of the arm; the *nut* is a shoulder near the small end of the shank, to prevent a wooden stock from slipping; the *pee*, or *bill*, is the point of an arm.

C is the *stock*, which stands at right angles to the shank and to the plane of the arms, and can be removed when required. Its usual length is equal to that of the shank added to half the diameter of the ring. When made of iron (as in Figs. 5 and 6), it is a round rod, and passes through a hole in the shank, near the eye for the bolt of the shackle. When made of wood (as in Plate $\frac{B}{5}$), it is divided lengthwise into two pieces, which are placed one at each side of the square part at the small end of the shank, and fastened together with four bolts near the shank, six or eight treenails, and four or six hoops, two of which are at the ends. This stock is square in section; its dimensions for the middle sixth of its length are equal to one-twelfth part of its length, and it tapers each way to one-half of those dimensions at the ends. The use of the stock is to make the anchor *cant*, or turn over, on reaching the ground, so that one or other of the flukes shall be sure to take hold.

According to the report of a committee which made an experimental comparison of several different anchors in 1852, the following are the qualities which a good anchor ought to have, with numbers affixed indicating their relative importance:—

Canting quickly.....	15
Holding on well.....	80
Strength of form and material.....	15
Exemption from fouling.....	10
Quick tripping.....	5
Ease of fishing in a heavy sea-way.....	10
Facility of stowing.....	10
Facility of sweeping.....	5
Ease of transport in boats.....	5
Sum.....	155

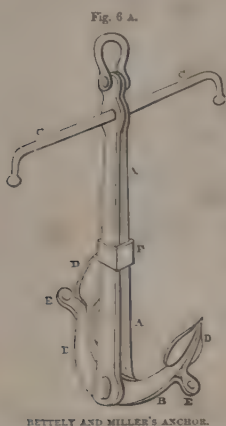
It is difficult, however, to see why the qualities of holding on and of strength should have been estimated at values so different as 80 and 15; for neither of those two qualities is of any use without the other.

Figs. 5 A and 6 represent the two anchors which, according to the report of the committee, stood highest as to general merit,

viz., first, Trotman's Anchor, Fig. 6; second, Rodger's Anchor, Fig. 5 A.

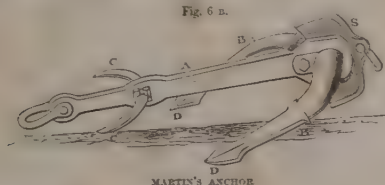
The advantages of the hinged anchor are, that it avoids the difficulty which is found in obtaining a sound forging at the crown of the common anchor; and that it is less liable to be fouled by the cable than the common anchor—that is to say, less liable to catch or entangle the cable with its upper arm; because when the lower arm has taken hold of the ground, the bill of the upper arm lies close to the shank.

Fig. 6 A represents a form of hinged anchor differing in some details from Fig. 6. The shank, A A, is divided longitudinally into two pieces, which are bound together by the square hoop, F, and against that hoop the bill of the upper arm presses. The spurs or horns, E, of the arms are made with eyes in them, to one or other of which the fish-tackle is hooked when the anchor is to be fished.



either way, in a hole in the shank. The cross-piece or sector marked S, forged upon the arms at the crown, serves at once

Fig. 6 B is a form of anchor, said by seamen to be very efficient, in which both flukes take hold of the ground at the same time. The palms and arms are in one plane, and they turn through an angle of about 40°



to limit the angle they make with the shank, and to cause them to take hold of the ground quickly.

On the subject of anchors, besides the report of the committee already referred to, reference may be made to "A Treatise on Ships' Anchors," by Mr. George Cotsell, N.A.

72. Cables.—The strength of cables, whether iron or hempen, has been considered in the Third Division, Article 87; and their usual lengths and number have been stated in the Appendix to that Division.

Iron chain-cables are commonly made in lengths of from 12½ to 25 fathoms (but the term *cable's length*, when used as a measure of distance, means 100 fathoms of 6·08 feet each, being one-tenth of a nautical mile). According to the rules formerly observed in the British navy, each length of 12½ fathoms had a *swivel* in it, to prevent twisting; but, by a recent Admiralty order, all ships having Brown & Harfield's capstans have two swivels only on each cable, one at either end: the swivels having been found to work unsatisfactorily round those capstans.

The several lengths of chain are joined together by means of *clackles*, sometimes called *joining shackles*, in order to distinguish them from the *anchor-shackle*, which fastens the cable to the anchor. A joining shackle is U-shaped, with the curved end pointing outboard; it is fastened with a bolt; and the bolt does not project beyond the eyes of the shackle, but is secured

with a small pin passing through both the bolt and the eye. The pin is fixed in its hole with a pellet of lead.

Chain-cables, when long and heavy, are stowed in compartments of the hold called *chain-lockers*. The nearer these are to the middle of the ship's length, the better is their position as regards liveness in pitching, to which heavy weights near the ends of the vessel are unfavourable. Accordingly, in sailing ships of the Royal Navy, the chain-lockers are near the main-mast; and in steamers, immediately before the engine and boiler compartment; but in merchant vessels they are often placed further forward with a view to convenience (as in Plate F). In river steamers and other small vessels, the chain-lockers are often boxes on deck, running on four small wheels.

The space required for the stowage of 100 fathoms of chain-cable may be computed approximately by the following rule—*Multiply the square of the diameter of the cable iron in inches, by 35; the product will be the space required in cubic feet, nearly.*

To find the riding-scope, or length of chain-cable, that should be payed out in order that it may lie horizontally where it is shackled to the anchor. Reduce the greatest working pull on the anchor to an equivalent length of chain-cable, weighed in water: call this length the *modulus*. To the modulus add the depth of water; from the square of their sum subtract the square of the modulus; the square root of the remainder will be the scope required.

From the data given in Article 87 of the Third Division, it appears that the weight *in air* of 100 fathoms of chain cable is ·135 of the test-load of the cable, or ·54 of the greatest working pull on the anchor. Deducting $\frac{2}{15}$ for loss of weight in water, the weight of 100 fathoms of chain-cable *in water* is found to be ·468 of the working pull on the anchor; hence the *modulus* to be used in the preceding calculation is,

$$\frac{100}{\cdot 468} = 214 \text{ fathoms, nearly.}$$

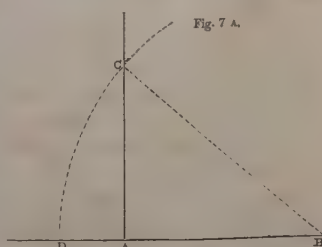
For example, let the depth of water be 40 fathoms; then—

	Fathoms.
To the modulus,.....	214
Add the depth,.....	40
Sum,.....	254
Square of the sum,.....	64,516
Subtract square of the modulus,.....	45,796
Remainder,.....	18,720
Square root of the remainder, nearly.....	137

In moderate depths, the scope of cable required varies nearly as the square root of the depth. The following are some examples of the results of the rule, calculated to the nearest whole fathom:—

	FATHOMS.									
Depth,.....	5	10	15	20	25	30	35	40	45	50
Scope,.....	47	67	82	95	107	118	128	137	146	155

To solve the same question graphically, draw (in Fig. 7 A) a straight line, D A B, and another straight line, A C, meeting the first straight line at right angles in the point, A. Then from A set off A B to represent the modulus, and A D, in the opposite direction, to represent the depth of water. About



B, with the radius, \overline{BD} , describe a circular arc, cutting AC in C; \overline{AC} will represent the required scope of cable.*

The scope in practice is seldom so great as that given by the preceding rule.

Hempen cables are large ropes, of the kind called *cable-laid*; that is to say, the several parts of which the thickness of the rope consists are spun, or *laid up*, at four successive stages, in contrary directions alternately, as in the following example:—

Hemp is laid up *right-handed* into yarns;
Yarns are laid up *left-handed* into strands;
Three strands laid up *right-handed* make a hawser;
Three hawsers laid up *left-handed* make a cable.

Hempen cables are stowed by being coiled in the *cable-tiers*, which are placed on the orlop-deck.

73. *Manger*.—The hawse-holes, with their hawse-pipes, bolsters, and plugs, have already been mentioned in Article 44 of this Division. In ships of war and large merchant ships they are usually four in number; the foremost pair being for the bower cables, and the aftermost pair for the sheet cables: small merchant ships have usually two only.

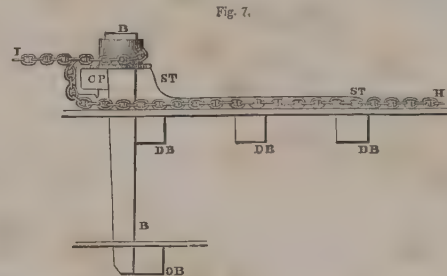
A short way abaft the hawse-holes, on the working deck, is a low upright partition, composed of planks lying athwartships, called the *manger-boards*, to prevent the water that comes in at the hawse-holes from flooding the rest of the deck. The triangular space before the manger-boards is called the *manger*; and a pair of scupper-holes for discharging the water at its after corners, the *manger-scuppers*. The ends of the manger-boards fit into rabbets in upright pieces called the *manger-stanchions*, of which there are either two or four, according as the manger-boards are in one or in three lengths. The manger-boards can be removed when the flat of the deck requires to be caulked or repaired. They are now little used in merchant vessels; and in many ships they are rendered unnecessary by the hawse-pipes being made to slope upwards from the hawse-holes, and so to conduct the cables to the deck next above the hawse-holes, instead of that next below.

74. *Controllers—Bits—Stoppers—Compressors*.—For the purpose of regulating and checking the motion of the cable as it runs towards the hawse-holes while the anchor is dropping, and also of holding on by the cable after the anchor has taken hold, four kinds of apparatus are used, together or separately—controllers, bits, stoppers, and compressors.

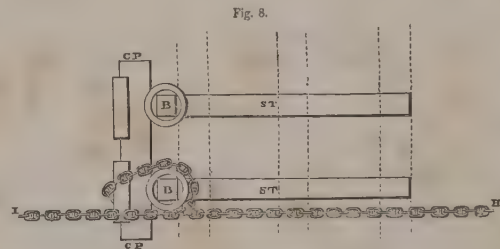
A *controller* is a cast-iron block, having a hollow in its upper side of the shape of a link of the chain-cable. Controllers are bolted to the deck at various points in the lines along which the cables lie on their way from the chain-lockers to the hawse-holes. A cable while lying on a controller tends of itself to drop into the hollow; and while there, it is held by one of its links, which lies flat in the hollow; but at the bottom of a hollow

is the short arm of a lever, which can be raised by hauling up the long arm, so as to lift the cable out of the hollow when required, and allow it to run.

The *riding-bits*, whose strength has been already considered in Article 87 of the Third Division, bear in ordinary the principal part of the tension of the cables. Their usual station is between the foremast and mainmast; and there are two pairs—the foremost pair for ordinary use, and the aftermost to be used in case the foremost pair should give way. Riding-bits are shown in some of the longitudinal sections and lower-deck plans given in the Plates. The two annexed figures show the arrangement and use of their principal parts—Fig. 7 being a



side elevation, and Fig. 8 a plan. DB, DB, DB, are lower-deck beams; OB, an orlop beam, under the aftermost of them. B, B are a pair of bits; being strong upright posts bolted



against the after-sides of the two last-mentioned beams, and connected together by means of the *cross-piece*, CP. ST, ST are two *standards*, or horizontal struts, abutting against the front of the two bits, and lying upon and fastened to three successive lower-deck beams, for the purpose of resisting the forward pull of the cables. HI is part of a bitted cable—H being towards the hawse-holes, and I towards the chain-locker. The bits may be made wholly of iron; but when of wood, the head of each bitt (which is square) is guarded with a strong thick cylindrical casing of cast-iron; and the back part of the cross-piece is similarly protected. Sometimes a pin is fixed in the upper side of the end of the cross-piece, to prevent the cable from slipping off. Some riding-bits have no cross-piece, but a large transverse pin instead of it. Other bits, of smaller size, are used in various parts of the vessel for securing different ropes: their general construction is the same as that of riding-bits, but on a smaller scale.

Deck-stoppers are short ropes or chains, shackled to bolts in the deck at one end, and secured at the other end to the cable by a fastening which can be *slipped*, or instantly let go, when required. The fastening commonly used for this purpose is called a *slip-hook*.

The *compressor* is usually a bent lever, shown in plan in

* When a ship, being in a confined anchorage, is compelled to ride at short scope, the cable might still be made to lie flat on the bottom at the anchor-shackle by loading it with additional weight (for example, by hooking pieces of chain to it), according to the following rules:—

I. From the square of the scope subtract the square of the depth of water: divide the remainder by twice that depth: the quotient will be the required modulus; that is, the length of cable, which, with its load, when weighed in water, should be equivalent to the horizontal pull on the anchor.

II. Divide the ordinary value of the modulus (say 214 fathoms) by the required value; the quotient will be the ratio which the gross weight of the loaded cable should bear to the weight of the cable alone.

EXAMPLE.—Suppose the depth to be 20 fathoms, and the scope 80 fathoms. Then—

$$\frac{80^2 - 20^2}{2 \times 20} = 150 \text{ fathoms, the required modulus; and}$$

$\frac{214}{150} = 1.43$ nearly, the ratio in which the gross weight of the cable should be increased by loading it.

Fig. 9, which moves horizontally close to the lower end of the *chain-pipe*, through which the cable comes up from the chain-locker to the lower deck.

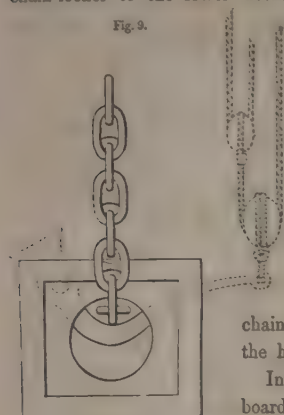


Fig. 9.

By hauling on the tackle at its end, it is made to press the cable firmly against the inside of the pipe, so as to moderate the speed of its running out. Another kind of compressor consists of a pair of jaws, with a screw like that of a vice for drawing them together, and holding the cable between them.

The usual diameter of the chain-pipes is two-thirds of that of the hawse-pipes.

Inside the chain-locker, the in-board end of the cable is secured by a *slip-hook*, so that it can be instantly let go when required.

75. *Cat-heads—Fish-davits—Bill-boards—Tumblers—Anchor-struts.*—When a *bower anchor* is weighed, and has been hove up clear of the water, it is hauled up to the ship's bow by means of a tackle called the *cat-fall*, which hangs from the *cat-head*, and is hooked for the time to the ring of the anchor; and the arms are afterwards hauled up till the shank lies nearly level, with the crown pointing aft, by means of another tackle called the *fish-fall*, which hangs from the *fish-davit*. The cat-head serves also to hang the ring of the anchor from, ready for letting go, by means of a rope called the *cat-head stopper*; while at the same time the other end, or throat, of the shank is hung at the same level, or nearly so, by a rope or chain called the *shank-painter*, and partly supported by the inner fluke resting on an iron plate or iron-covered board, called the *bill-board*, which projects from the side with a slight outward slope. The cat-head stopper and shank-painter are secured in-board by means of moveable pins called *tumblers*, which are acted upon by a lever that casts them both loose at one instant when the anchor is to be let go.

Examples of *cat-heads* are shown in several of the Plates, especially the sheer-plans and longitudinal sections of vessels, such as, $\frac{1}{2}$, $\frac{3}{4}$, &c. They are usually a pair of square wooden beams, one projecting from each bow, with a moderate *flight* or upward slope; and of a length sufficient to insure that the anchors shall hang from them clear of the ship's side. They are placed as far forward as may be convenient, and nearly on a level with the gunwale, or sometimes with the planksheer. The projecting part of each cat-head is supported by a knee called the *cat-head supporter* (of which examples are shown in the Plates), bolted to the cat-head and to the ship's side. The inner end of each cat-head (sometimes called the *cat's-tail*) is made fast either by returning down inside the ship's side, through which it is bolted to the supporter, or by lapping under a beam of the weather-deck or fore-castle, or by lying upon that deck and being bolted down to it; the last method being now the most frequent. The outer end of the cat-head is hooped, and has usually three mortises in it for the sheaves of the cat-fall. Cat-heads are sometimes of solid forged iron, and sometimes built of angle-irons and plates.

The *fish-davits* are a pair of davits, or small iron cranes, at

such a distance abaft the cat-heads as the length of the anchors may require.

The *sheet-anchors* are usually stowed immediately abaft the fore-channels (or projecting ledges for securing the rigging of the foremast, to be more fully described in the Fifth Division). The rings of the sheet-anchors point ahead, and rest on the after-ends of the fore-channels; the stock of each of them is upright; the shank lies horizontal; the inner arm rests on a projecting ledge called the *anchor-chock*; the anchor is secured by chain-stoppers round the stock and shank, which can be let go at once when required. Two sloping shores or *anchor-struts*, hinged to the ship's side below the anchor, abut against its shank; and when it is let go, they cause it to fall clear of the ship.⁹

76. *Capstans and Windlasses* are machines for winding up ropes and chains, and raising weights, as when an anchor is weighed. A capstan has its axis vertical, and is specially suited for being driven by hand-power, the men walking or running round it, and pushing before them the capstan-bars which radiate from its head. It is well calculated for making available the strength of a numerous crew. A windlass has its axis horizontal. When driven by hand, it is usually less powerful than a capstan, being worked by fewer hands; so that if it is to be made to lift the same load with a few men driving it that a capstan does with many, that can be effected only by means of mechanism, which diminishes the speed with which the load is lifted in the same proportion with the number of men. In order that a windlass may be equal or superior to a capstan, taking speed as well as load into account, it must in general be driven by steam-power; and this is much practised in merchant vessels.

A windlass for lifting goods is sometimes called a *crab*, or *winch*.

A large ship has usually two capstans, called the *fore* and *after* capstan respectively—the fore capstan standing midway, or nearly so, between the foremast and mainmast; the after capstan at about the same distance abaft the mainmast.

Capstans are distinguished into *single* and *double*, according as they have one or two barrels upon the same spindle, or vertical axis. The barrel of a single capstan, or the lower barrel of a double capstan, is on the deck on which the cables are worked, and is used for heaving in the cables; the upper barrel of a double capstan is on the deck above. In either case, the spindle has the framing of two decks to keep it steady; it turns in a bush or collar in the upper of those decks, and has the pivot at its lower end supported by a step fixed to the lower of them.

Fig. 10 is an elevation of a double capstan; and Fig. 11 a plan of its lower barrel.

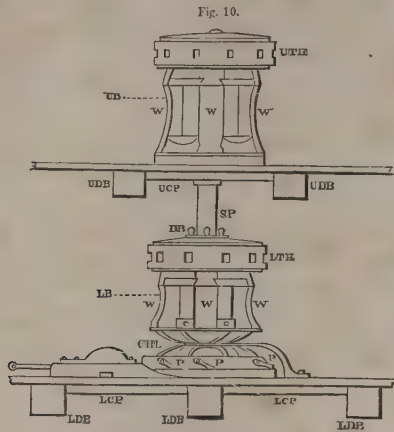
UCP are the upper, and LCP the lower *capstan-partners*, being strong platforms supported by the two decks, and composed, when made of wood, of pieces 6 or 7 inches deep, laid like carlings. UDB are upper, and LDB lower, deck beams.

SP is the *spindle*, of strong and tough wrought-iron. Its greatest diameter is about the middle, and ranges from 5 to 8 inches. It tapers towards the ends, where its diameter is about $\frac{2}{3}$ of the greatest diameter. When made of steel, its diameter may be reduced so as to preserve the same strength.

UB is the upper barrel, which is fast on the spindle, and

⁹ For details as to the working of anchors and cables, and the construction of various fittings connected with them, reference may be made to the work of Lieutenant Nares, R.N., on "Seamanship."

turns with it; and LB the lower barrel, which is loose on the spindle, but can be made fast to it when required, so as to turn along with the upper barrel, in the following manner:—On the top of the head of the lower barrel is a circular plate; and just above it, fixed to the spindle, a similar circular plate—these



are called the *connecting-plates*: they have corresponding holes in them; and by putting bolts, called *drop-bolts* (marked DB in Fig. 10), into those holes, the lower barrel is connected with the spindle, and made to turn with it.

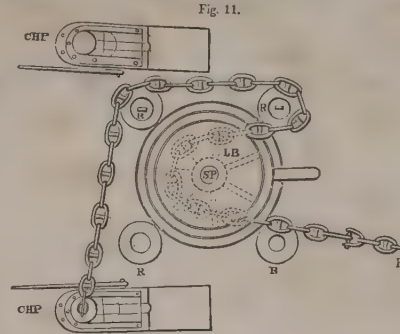
The barrel of a capstan ranges from 16 to 28 inches in diameter, and is usually polygonal, with ten or twelve equal sides. From the alternate sides ribs project, called *whelps* (W), which are consequently five or six in number; and they are of such a breadth that the mean diameter, measured over the whelps, is about double the diameter of the barrel. They are kept apart at their upper and lower ends by *chocks*. They taper towards the upper end; so that the diameter over the whelps is 4 or 5 inches less near the upper end than at the lower end; this is called the *surging power*; and its object is to make a rope, when wound round the capstan, gradually *surge*, or slip from the larger end, where it is led on, to the smaller end, where it is led off, in order that the successive turns of the rope may not override each other at the larger end.

UTH is the *trundle-head* or *drum* of the upper capstan, and LTH that of the lower capstan. The trundle-head of a capstan ranges from 3 to 5 feet in diameter, being a little greater in diameter than the greatest diameter over the whelps. It has square holes all round its outer rim for inserting the *capstan-bars*, at the rate of about one to every foot of its circumference, or nearly so; these *bar-holes* are from 3 to 5 inches square, tapering inwards, and from 10 to 12 inches deep. The length of the capstan-bars is about three times the diameter of the trundle-head, or from 8 to 14 feet long. They are fastened in the bar-holes by means of small pins dropped in from above, and are connected together all round by a rope through their outer ends, called a *swifter*.

P are the *pauls*, or catches, for preventing the capstan from running back. They are carried by a round part of the capstan called the *paul-head*; and they drop between and take hold of the teeth of the *paul-rim* or *ratchet*, a strong toothed ring which is let down into and bolted to the lower partners. When required, the pauls can be supported clear of the ratchet by means of small pins.

CHL is the *chain-lifter* or *cable-holder*, made of cast-iron, for acting directly on a chain-cable. Its rim is of the form of a deep groove, with projecting ribs on its upper and lower surfaces; so that the alternate links of a chain may fit into the spaces between the ribs.

R, R, R, R, in Fig. 11, are upright rollers, for guiding a chain-cable, so as to make the chain-lifter lay hold of it.



CHP, CHP, in Fig. 11, are a pair of *chain-pipes* or *deck-pipes*, with controllers or deck-stoppers just ahead of them. A cable is represented coming from the bows (which are in the direction marked H), guided round the capstan by the rollers, and dropping into the starboard chain-pipe.

This mode of fitting a capstan, so as to enable it to act directly on a chain-cable, is known as *Brown's fittings*. In the absence of such fittings, or when a hempen cable is to be hove in, an endless chain or rope, called the *messenger*, is used, passing round the capstan, and round two pulleys near the hawse-holes. If a chain, the messenger is acted on by a *sprocket-wheel*, having teeth suited to the size and figure of the links: if a rope, it is put three times round the capstan-barrel. The messenger lies alongside the cable, to which it is fastened by iron or rope fastenings called *nippers*; these are successively taken off the part of the cable that is approaching the capstan, and put upon the part that has just come in through the hawse-hole.

The average total power of a man working at a capstan-bar is estimated at about 50 foot-pounds per second, or 3000 foot-pounds per minute; being $\frac{1}{4}$ of a horse-power.

A *windlass* of the old form, employed in small vessels, consists mainly of a barrel with a horizontal spindle, turning in bearings supported by upright posts called the *earrick-bitts*; provided with whelps, and also with a ratchet-wheel and pauls to prevent its running back; and driven by means of *hand-spikes* inserted into holes in the barrel. Improved windlasses have chain-lifters for heaving in chain-cables, and gearing of a great variety of kinds, more or less like the wheel-work of a crane, for enabling a small force with a great speed to overcome a great resistance slowly. Sometimes also the windlass is provided with a friction-brake, to be used in lowering weights; and then if the brake is powerful enough, the anchor may be lowered by means of it, without the aid of bitts or compressors. A windlass may be driven either by a small steam-engine forming part of its own mechanism, or by a messenger or endless chain, from an engine used for various purposes and placed in any convenient part of the vessel.

The shape and position of the windlass render it a convenient machine both for heaving-up chain-cables, and for paying them out by the aid of a friction-strap. On the other hand, the shape and position of the capstan are peculiarly well suited for employing the strength of a large number of men. Those advantages may be combined by having a capstan on an upper-deck, driving a windlass on the deck next below by means of a bevel pinion on the spindle of the capstan, gearing with a bevel wheel on the windlass. This is done in Emerson and Walker's windlass, as to which, see Nares on "Seamanship;" and by means of two sets of pauls, the capstan is so connected with two bevel pinions driving bevel wheels of different sizes, that by turning the capstan in one direction or in the opposite direction, two different speeds can be given to the windlass.

77. *Ships' Boats*.—As to the styles in which boats are built, see Article 68 of this Division. The sizes and usual mode of stowage of the boats of a large ship of war are illustrated in the Upper-deck Plan, $\frac{B}{5}$. The names, styles of build, and usual lengths and proportions of length to breadth of the principal classes of ships' boats, are shown in the following table:—

SHIPS' BOATS.

CARVEL-BUILT OR DIAGONALLY-BUILT.

Name.	Length. Feet.	Length \div Breadth.	Remarks.
Launch,.....	from 34 to 42	... from $3\frac{1}{2}$ to 4.	{ Strong heavy flat-floored boat, 10 to 12 oars; sometimes carries a gun.
Long-boat,.....	do.	... do.	{ Like a launch, but sharper in the floor.
Barge,.....	from 30 to 32	... do.	{ 10 or 12 oars; boat of state for flag-officers and captains.
Pinnace,.....	from 28 to 32	... do.	{ 6 or 8 oars; boat for captains, commanders, and lieutenants.
Yawl,.....	from 23 to 28	... do.	Boat for ordinary use.

CLINKER-BUILT.

Galle ^r ,.....	from 28 to 36	... from $4\frac{1}{2}$ to 5.	{ 10 or 12 oars; light sharp boat, for speedy rowing on expeditions.
Gig,.....	from 22 to 24	... do.	{ 4, 6, or 8 oars; similar to a galley, but smaller.
Cutter,.....	from 22 to 30	... from $3\frac{1}{2}$ to 4.	{ For general use; like pinnace and yawl, but lighter as being clinker-built.
Jolly-boat,.....	from 16 to 20	... from 3 to $3\frac{1}{2}$.	{ Like a small cutter: for general use.
Dingy,*.....	from 12 to 14	... about 3.	For general use.

In addition to the boats referred to in the table, may be mentioned *troop-boats* for embarking and disembarking troops, and *paddle-box boats*, which are very broad and flat, and are made to fit bottom upwards on the tops of the paddle-boxes of a paddle steamer: these also are well adapted for landing troops.

The boats commonly used in merchant ships are long-boats, yawls, cutters, jolly-boats, and dingies; and *life-boats* are also often used, which are like cutters or jolly-boats, according to their size, but shaped alike at both ends, and with the addition of air-cases or cork floats running round both sides under the ends of the thwarts. For example, in the upper division of the upper figure of Plate $\frac{A}{3}$, there are seen two yawls slung at one of the vessel's quarters, and two life-boats at the wings of one of her paddle-boxes. In the lower figure of Plate $\frac{B}{5}$, a jolly-boat is slung at one quarter and a life-boat at the other.

* The "g" in "Dingy" is pronounced hard.

Boats not likely to be immediately wanted are usually stowed above the upper-deck in the waist of the ship, sometimes bottom up, on beams, as illustrated in Plate $\frac{F}{1}$, and sometimes upright, on crutches, as shown in Plate $\frac{B}{5}$. Boats for immediate use are slung each by means of a pair of *boat-tackles*, from a pair of *davits* or small cranes, usually shaped like that shown in Division Third, Article 54, Fig. 17. The boat has a pair of *slings*, being short ropes made fast to ring-bolts in the keelson, near the head and stern; from near the upper ends of the slings *steadying lines* pass to the sides of the boat to prevent it from canting or turning over; and at the upper ends of the slings are two hooks, hooking into *thimbles* or rings at the lower ends of the tackles. The hooks are on the slings, and not on the tackles, lest in letting go the tackles a man should be hooked out of the boat. The davits for pinnaces, yawls, cutters, galleys, &c., are usually at or near the ship's quarters, and the boats slung from them are called *quarter-boats*; the quarter-davits can be turned about so as to make the boat hang either outboard or inboard; from a pair of davits projecting over the stern is hung the *stern-boat*, which is usually a jolly-boat, gig, or dingy. The last-mentioned pair of davits are often simply a pair of straight projecting arms, like the cat-heads. Boats hanging from davits are shown in Plates $\frac{A}{3}$, $\frac{B}{5}$, and various other Plates. To keep a boat in such a position from swinging about, it is secured to the lower part of the davits by the *gripes*, which are a pair of bands passing round the boat near the head and stern. Each of the gripes (according to the best construction) has a ring or *thimble* at each end, which in securing the boat is slipped upwards on to a pin or *prong* pointing downwards, so that when the boat is lowered the thimbles slip down off the prongs of themselves, and cast the gripes loose. Each of the gripes can be set taut by means of a lanyard.

In order that a boat may be capable of being lowered according to *Clifford's method*, it must be provided with a horizontal thwartship roller or barrel amidships. Two ropes called *lowering pendants* hang from the davits, and are led below sheaves fixed to the floor of the boat, near its head and stern respectively, and thence to the barrel, upon which they are wound in the same direction, their ends passing loosely through holes in the barrel. A third rope called the *lowering-line* is made fast to the barrel, and is wound round it in the contrary direction,* so that being passed twice round a cleat on the boat's midship thwart, it enables one man who holds it, standing in the boat, to control the motion of the barrel, and let the boat descend gently and steadily into the water. To increase the friction which controls the descent, and also to keep the boat from canting over, the upright parts of the lowering pendants pass through two three-sheave blocks, at the upper ends of the slings of the boat, which are unhooked from the tackles before lowering.—(See Nares on "Seamanship.")

For the regulations in force in Britain as to the number and size of the boats with which passenger ships are to be supplied, see the Merchant Shipping Act.

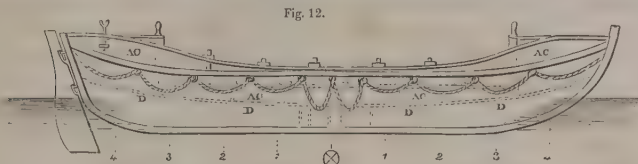
78. *Life-boats*.—The ordinary ships' life-boats referred to in the preceding Article are imperfect; because although the air-cases, or the floats which run round their sides, may enable

* In many of the published figures of Clifford's method of lowering boats, the lowering line is shown as wound upon the barrel the wrong way.

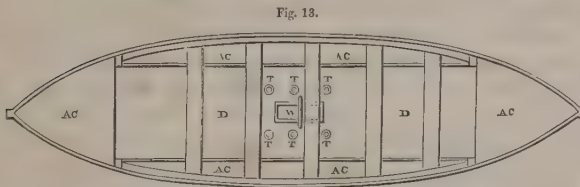
them to float when full of water, they want the power of *self-righting*, which is essential to a complete life-boat.

In an ordinary boat, the inverted position is a position of stability, as well as the upright position; and should the boat upset, it continues to float bottom upwards. In order to make a boat *self-righting*, the inverted position must be rendered unstable; and that is done by giving the boat a great sheer upwards at the bow and stern, and filling the ends which are thus raised with large buoyant air-cases. In some life-boats the side air-cases are carried up to the gunwale for the same purpose.

Figs. 12, 13, 14, and 15 show the general construction and arrangement of one of the life-boats of the Royal National Life-



boat Institution, on a scale of $\frac{1}{80}$ of the real dimensions. Fig. 12 is the sheer-plan, Fig. 13 the deck-plan, Fig. 14 the body plan, and Fig. 15 the midship section. The length of



the boat represented is about 30 feet; the breadth $7\frac{1}{2}$ feet, being one-fourth of the length; the depth amidships is about 3 feet; and at each end there is an upward sheer of about $2\frac{1}{2}$

feet. The lower gunwale has a regular upward curvature; and above it is an upper gunwale or plank-sheer, rising rapidly with a reverse curve, as Figs. 12 and 14 show. The midship section is nearly rectangular, and flat-bottomed; towards the ends, the cross-sections become U-shaped, as shown in Fig. 14, with a slight flare out. There is a heavy iron keel, which acts as ballast.



In Figs. 12, 13, and 15, D is the deck, having a moderate upward sheer: the space below it, for a length of rather more than one-fifth of the boat's length amidships, is filled up solid with light wood, marked F in Fig. 15. Amidships is a small well, W, with a pump for removing water which may leak into the space below the deck.

T, T, T, T, T, T, are six *relieving tubes*, 6 inches in diameter, with valves opening downwards, for discharging any water which may lodge on the deck.

AC are the air-cases of the sides and ends. The side air-cases are below the ends of the thwarts; the end air-cases rise to the level of the heads of the stem and stern-post respectively.

In Fig. 12 are shown the *life-lines*, hanging in festoons or bights round the sides of the boats, for persons in the water

to hold on by. The two midship bights at each side hang lower than the rest, so as to form stirrups to assist men in clambering on board.

Life-belts or jackets made of cork should be provided for the whole crew of the life-boat.

79. *Life-buoys*.—The life-buoy for hanging over the stern, with which ships of war, and sometimes also merchant ships are fitted, is usually made of copper or yellow metal, and consists of an upright shank with a pair of horizontal arms, forming a cross, and having at the ends of the arms two thin hollow globes, to give buoyancy. At the lower end of the shank is a flat step or stirrup, for the foot of a man to rest upon while he holds by the arms, or by the lower part of the shank.

Should he take hold of the upper part of the shank, the buoy capsizes. On the top of the shank is a port-fire, to show the position of the buoy at night: it is lighted by a trigger, which is pulled before the buoy is let go.

Circular life-buoys are large rings, filled with cork and covered with painted canvas, and having ropes round them to take hold by. They are distributed about the weather-decks.

80. *Pumps*, for discharging water from the ship's hold, usually stand in compartments called *pump-wells*, which extend from the ship's bottom to the lower-deck, and sometimes to the upper-deck. (See, for example, Plate F, where a pair of pumps are shown, immediately abaft the mainmast.) In iron ships, divided into compartments by water-tight bulkheads, there is usually a pump, from 6 to 8 inches in diameter, to each such compartment. Pumps draw the water from the *limbers*, or water-channels (already mentioned in this Division, Article 38), and discharge it into the sea through a channel or spout called the *pump-dale*. Pumps were once made of wood, but are now generally of mixed metal, or of iron. They are of an endless variety of kinds, both as to their own construction and that of the mechanism by means of which they are worked; but for the most part they belong to one or other of the three following classes:—

I. *Piston-pumps* are the most frequently used; and are those in which a piston either moves up and down or to and fro in a cylinder or barrel, or revolves in a circular casing—the former being by far the more usual form. Every pump with a reciprocating piston requires at least two clacks, or self-acting valves opening upwards, for the supply and discharge of the water respectively; the casings or chambers which contain those valves are called respectively the *lower* and the *upper pump-box*. When the lower pump-box is fixed, and the upper pump-box is also the piston, the pump is called a *sucking-pump*; and this is the oldest construction: when the lower pump-box is the piston, and the upper is fixed, the pump is called a *lifting-pump*; this construction is not usual on board ship: when both pump-boxes are fixed, and the piston is solid, the pump is called a *forcing-pump*; and this is the most efficient construction. A long solid piston, without packing, is called a *plunger*. The piston-rod is called by seamen the *pump-spear*. In a *double-acting forcing-pump*, there are two sets of pump-boxes, connected with the two ends of the cylinder or barrel, so that the piston may force up water during the return-stroke as well as during the forward-stroke; in this case the piston-rod must pass through a stuffing-box.

Reciprocating pumps may be worked either by means of levers (called by seamen *pump-brakes*), or by means of cranks on a revolving shaft.

II. The *Chain-pump* consists of a tube with its lower end dipping into the limber, and of an endless chain passing up the tube, over a sprocket-wheel, and down another tube called the *back-casing*, and carrying a series of circular discs which nearly fit the tube, but not so closely as to rub against it. The sprocket-wheel is turned by means of cranks or winches on its axle, and the discs are thus made to drive the water before them up the tube. At the upper end of the tube is a cistern, whence the water flows away by the pump-dale.

III. The *Centrifugal pump* consists of a fan-wheel rotating within a circular casing, at a speed sufficient to produce in the mass of water within the casing an outward pressure intense enough to raise it from the level of the limbers to that of the pump-dale, and discharge it overboard. The water is drawn into the casing at the centre, through holes at both sides, and discharged at the circumference, either through one pipe or through variously formed passages. In the most efficient centrifugal pumps, the vanes of the fan are curved backwards, in order that their leading edges may cleave the water without striking it; and outside the circumference of the fan, there is sufficient space to allow the rapid motion at first impressed on the particles of water by the vanes to subside, and to be replaced by pressure.

In all sorts of pumps it is essential to economy of power that the passages traversed by the water should be as roomy, as short, and as direct as the circumstances of the case will admit of, and should be free from sudden contractions, sudden enlargements, and sharp turns; and this requires special attention where there are valves.

The *efficiency* of good pumps, or proportion of useful work to total work, may be estimated as ranging from $\frac{2}{3}$ to $\frac{3}{4}$.

The *work of a man* in pumping, when the exertion is kept up for *eight hours per day*, may be estimated as equivalent nearly to an effort of $17\frac{1}{2}$ lbs., exerted through $2\frac{1}{2}$ feet in each second, or 150 feet per minute, or 9000 feet per hour; being 157,500 foot-lbs., or 70 foot-tons per hour; or 560 foot-tons per day of eight hours. But a much greater exertion than this can be kept up for a few minutes at a time.

Experience in the working of *fire-engines* has shown, that the most favourable length of stroke of pump-handles to be worked by hand-power is from 30 to 35 inches. A stroke of 42 inches can be worked by men specially trained to it, but is too long for other men in general. The number of men required to work a hand fire-engine of the best kind, is about *one man for every 22 cubic inches of pump-barrel* (or in other words, *one man for every 28 cylindrical inches*): the contents of a pump-barrel being found by multiplying the area of the piston by the length of its stroke. If the piston is double-acting, this is to be doubled. When the preceding conditions are observed, strong active men, if frequently relieved, can work fire-engines at the rate of nearly 60 effective strokes per minute.

A portable fire-engine usually has a pair of single-acting forcing pumps of about 8 inches length of stroke, acting alternately, and forcing water into an air-vessel, whence it comes out in a continuous stream. The area of piston is adapted to the number of men available to work the engines. The ordinary

diameter of the hose, or flexible pipes for taking in and discharging water, are, for suction-hose, 3 inches or thereabouts; for delivery-hose, from 2 to 3 inches: the diameter of the nozzle ranges from 0.6 to 0.8 inch.*

Pumps may be driven by steam-power, supplied either by small engines for the purpose, or by the engines that propel the vessel. It is usual to fit every steam-boat engine with one or two *bilge-pumps*, for discharging water from the hold. When a steam-vessel leaks very rapidly, the water may be discharged by opening a valve from the hold into the condenser: the air-pump of the engine then becomes available as a bilge-pump; but this expedient is not to be used except in cases of emergency, as the foul water from the hold is injurious to the engine.

In some cases pumps have been worked by means of the pitching of the vessel and the heaving of the waves, by setting a loaded cask to float astern, and leading a rope from it by means of pulleys to the pump-spear.

81. *Tanks*, for holding a store of fresh water, are built of iron plates (which ought to be galvanized), and are usually rectangular in plan, and 4 feet square, or thereabouts. They are from 4 to 6 feet deep. They hold from 400 to 600 gallons. A *gallon* is $\frac{1}{1604}$ of a cubic foot, and, when the water is pure, weighs 10 lbs. Tanks are flat-topped, and most of them are also flat-bottomed; but some have one of the lower edges of the base tapered off, that they may fit into the bilge of the ship: these are called *bilge-tanks*. Each tank has a man-hole at one corner of the top, with a cover to fit it; and they are stowed so as to bring four man-holes together. They are stowed at the bottom of the hold, on a skeleton-floor, and arranged so as to bring their tops as nearly as possible to one level, the tallest tanks being placed amidships. (See Nares on "Seamanship.") In almost all the hold plans and longitudinal sections of ships given in the Plates of this Treatise, the water-tanks are shown.

It is advisable, on account of risk of fire, that the *spirit-room* should be a tight iron tank, entered only through a hatch in the top: care being taken to ventilate it properly, lest those who enter it should be stupefied or suffocated by the fumes from the casks and bottles.

82. *Ventilators*.—Ships are usually ventilated by guiding or forcing fresh air down into the places where it is required. The oldest contrivance for that purpose is the *wind-sail*, being a large tube of canvas, kept open by hoops inside, and slung by ropes in a vertical position down a hatchway. The top of a wind-sail forms a hood with a large vertical opening, which is directed to windward. Fixed wind-sails, or ventilating tubes of sheet copper, brass, or iron, are also used, passing vertically downwards through the decks to the space to be ventilated, and having bell-mouthed hoods at top, which are turned to windward. In almost all the vertical sections of ships given in the Plates, several such ventilators are shown. They are especially required in the engine-room and stoke-hole of a steamer.

To promote the circulation of air in the cabins and sleeping-berths of a ship, it is useful to make the bulkheads or partitions which inclose them pervious to air, though not to light. One way of doing this is to make each panel of a bulkhead consist of two layers, being a pair of wooden gratings with their bars

* See Report of the Special Jury on Fire-Engines, in the Reports of the Juries upon the International Exhibition of 1862. As to pumps in general, see Mr. D. K. Clark's work on the Exhibited Machinery of 1862.

standing obliquely in opposite directions, so as to form a set of angular passages through the bulkhead of such a shape that they cannot be seen through. This is specially useful in hot climates. In all climates it is advisable that all bulkheads between decks should have apertures for ventilation at the top and bottom.

Foul air may be discharged through chimneys in any convenient position; their tops should always rise high above the level of the mouths of the wind-sails. Hollow iron and steel masts, and pipes leading into the funnels of steamers are sometimes used as chimneys for the discharge of foul air.

Where the construction of the ship is such as to make ordinary means of ventilation insufficient, fresh air is distributed, and foul air drawn off, by means of blowing-fans driven by steam-power, as shown in the various plans and sections of the inboard works of H.M.S. *Warrior*, Plate $\frac{2}{5}$, &c. In order that a blowing-fan may work with economy of power, and with as little noise as possible, the same conditions should be fulfilled as in a centrifugal pump: viz., the blades of the fan should cleave and not strike the air, and space should be given beyond the circumference of the fan for the violent motion at first impressed on the air to subside by degrees.

The thorough ventilation of every part of a ship is essential to the preservation of the timber from dry-rot, as well as to the health of those on board.

Foul air chimneys and stove chimneys should have hoods pointing to leeward.

According to the information collected by General Morin in his work on Ventilation, the supply of fresh air introduced into really well-ventilated places where large numbers of persons are assembled, varies from 0.4 cubic foot to 0.8 cubic foot per head per second, averaging about 0.6 cubic foot per head per second, except where there is some special cause of insalubrity (as in hospitals, and in places where unhealthy trades are carried on), and then it may be necessary to increase the supply per head per second to 1 cubic foot, or sometimes to 1.5 cubic foot.

The sizes of the openings and passages for admitting fresh air and taking away foul air are regulated by the quantity of air, and by the velocities which experience has shown to be the most proper for the current of air. According to the same authority, the following are the best velocities for the air in different positions:—

	Feet per Second.
At the outlets, or orifices where foul air escapes from a room, from.....	2.5 to 3.3
At the inlets, or orifices where fresh air enters a room, from.....	
In tubes, trunks, chimneys, and other passages for fresh or foul air, about.....	1.3 to 1.6 "
	12.

Fresh air distributes itself in the most uniform manner throughout a room when it is introduced at a number of openings at as high a level as possible.

When a fire is used, not to warm a room, but simply to produce a draught in a foul-air chimney, the area of fire-grate may be about $\frac{1}{20}$ of the area of the chimney; and the consumption of fuel should be at the rate of about 1 lb. of coal to each 20,000 cubic feet of air.

The supply of air required for steam-boiler furnaces will be considered in the Sixth Division.

* The velocity of the fresh air entering is thus restricted in order that the draught may not be unpleasant to the inmates of the rooms.

83. *Warming*.—The heat of a fire is given out partly by radiation from the glowing fuel, and partly by conduction from the hot gases produced by the combustion. In the case of coal, about one-half of the heat is given out by radiation, and one-half by conduction; therefore if a cabin or other room is warmed by means of an open fire-place, so constructed that the radiant heat alone is made available, about one-half of the whole heat produced is wasted. Hence aboard ship, for the sake of economy of fuel, some kind of stove is commonly used. As warming by direct radiation, however, is more healthful than warming by conduction, a stove open in front, so that as much heat as possible shall be radiated, leaving that heat only which would otherwise be wasted to act by conduction, is preferable to a close stove, in which the radiant heat is absorbed by the metal of the stove, and afterwards given out by conduction to the surrounding air. To make the conducting surface act very efficiently in warming air, its extent should be about one square foot for each cubic foot per second of air to be warmed.

The most wholesome metal for the heating-surface of a stove is iron; for copper and brass, when hot, give out noxious fumes.

The expenditure of fuel required for thoroughly warming air in cold weather, may be roughly estimated at about 1 lb. of coal for each 3300 cubic feet of air.

84. *Water-supply*.—Salt water from the sea, or fresh water from a tank, can be supplied to any part of the ship where it may be required, by means of pumps and pipes, which need no special explanation. Water for cleansing the hold is admitted directly from the sea, when required, through the *sweetening-cock*.

Distilled water is obtained by condensing steam in a surface condenser. On board a steam-vessel, the steam may be taken from the engine boilers; in a sailing vessel a special boiler is required; and then, to promote economy of fuel, the water for feeding the boiler should be that which has been heated by the condensation of the steam in the surface condenser.

Distilled water, when condensed without proper precautions, is nauseous and unwholesome, owing to the presence of impurities, and the absence of the air which good water contains in a state of diffusion. The air diffused in good water contains proportionally more oxygen, and more carbonic acid, than the atmospheric air. In the apparatus of Dr. Normandy, the aëration of the distilled water is insured by retaining amongst the steam while it is in the act of condensing, not only the air which is disengaged from the water that is evaporated, but also the air that escapes from the whole of the water used for condensation: that additional quantity of air being necessary, because sea-water contains proportionally much less diffused air than good fresh water from lakes, springs, or rivers. The condensed water being thus well aërated, is filtered through animal charcoal to remove impurities, and at the end of the process is as good as that of the purest springs. In the process of Messrs. Chaplin, a supply of air is drawn into the condenser from the atmosphere. (See Jury Reports on the International Exhibition of 1862; also, Mr. D. K. Clark's "Exhibited Machinery of 1862.")

Mere filtration will not remove organic impurities from water, unless it has been first aërated.

Distilled water may be obtained from the jackets of the cylinders of a steam-vessel while the engines are working. To make it fit to be drunk, it should be aërated and filtered. In the screw-steamer *Lancefield*, already mentioned, the distilled

water from the jackets is distributed by pipes to the state-rooms, for purposes of washing.

85. The *Galley*, *Caboose*, or *Cook-room*, is a room, usually in a house on the upper-deck, containing the cooking apparatus. Various examples of its position are shown in the Plates. It is built either of iron or of wood. If of wood, the wood-work is usually lined with lead, having a layer of felt between the lead and the wood. The floor is often paved with fire-clay tiles. Care should be taken to ventilate the cook-room well.

85A. *Water-closets*, in ships of war, are often placed out-board, those for the seamen being on a grated platform at each side of the knee of the head, screened from view by the *berthing-boards*, or planking of the head-rails, and having vertical metal soil-pipes of uniform diameter descending through the cheeks of the head; while those for the officers are in the quarter-galleries. For an example of the former position, see the upper-deck plan of H.M.S. *Warrior*, Plate 5. Another very common out-board position in paddle-steamers, is on the *wings*, or grated projecting platforms, before and abaft the paddle-boxes. Various positions in-board for water-closets are illustrated in the Plates of deck-plans and longitudinal sections. The soil-pipes of in-board water-closets are usually about $2\frac{1}{2}$ inches in diameter, and have outlets into the sea so placed as to be covered with water when the ship is pitching and rolling in a seaway. They are often made with slide-valves; because the action of a common valve is liable to be deranged by the pitching of the vessel. It is advisable that every in-board water-closet should be ventilated through independent passages or openings, having no connection with the adjoining cabins or other rooms. Water-closets below the water-line have the soil removed and discharged by means of a pump.

86. *Lightning-conductors* are used to protect ships against the destructive effects of electric discharges between the sea and the clouds. The electric discharge takes place between two bodies in opposite electrical states, along the line of least resistance; and its destructive effect is greater, the greater the resistance with which it meets. The objects of a conductor are, to establish a definite line of least resistance, along which every discharge within a given space is certain to take place; and to insure that the resistance of that line shall be so small that no destructive effects shall arise from any discharge along it. Lightning-conductors are required on each mast of a ship with wooden masts, and on the jib-boom and bowsprit.

It is essential that every lightning-conductor should have a sufficient sectional area, otherwise it may be melted by a flash; that it should form an unbroken metallic communication from the mast-head to the sea, for at every break in a conductor, an explosion may take place; and that its course should be as nearly as possible straight, because if its course is indirect, it may cease to be the line of least resistance.

The metal which has the greatest conducting power for electricity, or in other words, offers the least resistance to it, is copper; and to make an efficient lightning-rod, the sectional area of copper should be not less than from $\frac{1}{4}$ to $\frac{1}{2}$ of a square inch. To give an iron rod equal conducting power, its sectional area should be about three times as great; and the iron should be pure and soft, and may be protected by galvanizing it.

According to the system introduced by Sir William Snow Harris into the Royal Navy, a conductor is made in the form of

a double strip of copper, sunk into a groove in the after side of each mast, and the lower side of the jib-boom and bowsprit. The two strips break joint with each other. As each mast consists of pieces, of which the lowest, or lower mast, is alone fixed, while the upper, or top-mast and top-gallant-mast, are capable of being raised and lowered by sliding through pieces called caps, those caps are provided with tumblers, by means of which the connection between the divisions of the conductor is kept unbroken. The same arrangement is made to connect together the divisions of the conductor on the jib-boom and bowsprit. In wooden ships the lower end of the conductor of the bowsprit runs down the stem, and is connected with the copper sheathing; and the lower ends of the conductors of the masts are also connected with the sheathing by means of bolts passing through the ship's bottom. In iron ships it is sufficient to connect the conductors with the iron hull of the ship. When the conductors are of copper, care should be taken to connect them with the iron of the ship at points as little as possible exposed to wet, and at the same time easily accessible for the purpose of seeing whether the iron is corroded through galvanic action. A piece of zinc in close contact with the iron where the copper conductor joins it, tends to prevent that corrosion.

Iron or steel masts are themselves conductors, and render copper conductors unnecessary. If used in wooden or composite ships, care should be taken that they have an unbroken metallic communication with the water by means of galvanized iron bolts.

Wire-rope standing rigging answers for a conductor when it is straight and taut, and has metallic communication with the sea; but when a shroud or a backstay hangs in a bight, it may cease to form a line of least resistance, and so may cause an explosion.

Instead of conductors made of rods, ships are sometimes protected by means of copper wire ropes, one hanging from each mast-head. The lower ends of those ropes are in ordinary kept coiled up on the weather-deck; and when a thunderstorm is expected, they are cast loose and dropped over the side, so as to dip into the water.

A lightning conductor protects but a small space around it; according to Sir William Snow Harris, that space may be considered as bounded by a cone having its apex at the top of the conductor, and its base of a radius equal to twice the height of the conductor; and hence the necessity for having a conductor on each mast; because a conductor on one of the masts does not protect the others.

87. *Lights*.—According to regulations enforced by the British and French governments, every vessel under way after sunset and before sunrise is to carry lights, as follows:—

A green lamp on the starboard side, and a red lamp on the port side, each visible for a distance of at least two miles on a dark night and in a clear atmosphere, and each visible throughout an arc of 10 points, from right ahead to 2 points abaft the beam on its own side of the ship; but neither of those side-lamps is to be visible from the contrary side of the ship; and to prevent that, each of them is to have an in-board screen projecting at least three feet forward from the light:—

Sea-going steam-ships (but not sailing ships), in addition to the side-lamps, are to carry a white light at the foremast-head, visible at least five miles off in a dark night and clear atmo-

sphere, and throughout an arc of 20 points, extending from 2 points abaft the beam on the starboard side to 2 points abaft the beam on the port side:—

Steam-vessels towing other vessels are to carry two such foremast-head lights vertically, in addition to the side-lamps:—

Vessels at anchor are to carry, at a height not exceeding 20 feet above the hull, a white light, visible at least one mile off all round the horizon.†

88. *Binnacles, &c.*—The construction and adjustment of compasses, and the correction of the errors caused in them by the iron of the ship, form a subject that can be explained in a satisfactory manner in a special treatise only; and it will therefore be not here further mentioned, except by referring for information regarding it to the "Admiralty Manual for ascertaining and applying the Deviations of the Compass caused by the Iron in a Ship," edited by F. J. Evans, Esq., R.N., F.R.S., and Archibald

Smith, Esq., M.A., F.R.S. The steering-compasses are placed in strong metal or wooden boxes called *Binnacles* or *Bitacles*, of which a well-appointed ship has two or three. They have openings glazed with strong plate-glass, to see the compass through, and at night are lighted by lamps inside. They are securely fastened to the deck.

A *hanging* compass is one so fitted that the card can be seen and read from below. It is sometimes placed in the top of the commander's cabin, so that he can see, without quitting his cabin, which way the ship's head bears.

As to the smallest admissible equipment of compasses, see the "Rules of the Liverpool Registry."

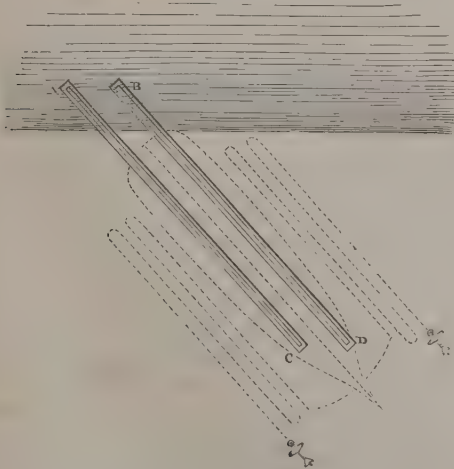
The *belfry* usually consists mainly of a pair of bitts or upright supports, between which the ship's bell swings. Its ordinary positions are exemplified in such of the Plates as show longitudinal sections of vessels.

CHAPTER VI.

LAUNCHING.

89. The *Launch* is a term used to comprehend the whole apparatus for launching the ship, together with the slip on which she is built and its equipments. The building-slip, with

Fig. 1.



its blocks, &c., has been described in Article 57 of this Division; so that the apparatus specially connected with launching remains to be described. It may be divided into two principal parts:

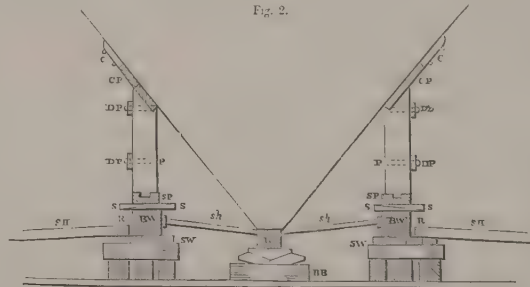
* The inboard screens, as prescribed by the regulations, are by no means sufficient to insure that the side-lamps shall not be seen across the bow, unless those lamps are provided with proper lenses, which consist of a semicircular part forming the inboard half of the front of each lantern, and a cylindrical part extending from the centre of the front of the lantern, round the outboard side, to two points abaft the beam. Their effect is to concentrate the light in a horizontal layer, and to confine it to the required angular space. The mast-head light should have a cylindrical lens. The best lenses are of the form called "polyzonal," but may be cast in one piece. The colouring of the side-lights is usually effected by means of coloured glass chimneys. As red glass or other colouring medium absorbs more light than a green medium of equal depth of colour, it is advisable that the flame of the port side-lamp should be somewhat larger than that of the starboard side-lamp, in order that both may be visible with equal clearness at the same distance.

† For details regarding pilot-vessels, fishing-boats, &c., see the Regulations as issued by the Board of Trade.

the *sliding-ways* or *slip-ways*, which rest on the floor of the slip, and present a smooth upper surface; and the *cradle*, being a temporary framework which rests and slides upon the slip-ways, and supports the ship during the launch.

In the sketch plan, Fig. 1, AC and BD are a pair of slip-ways, and the dotted outline marks the position of the ship. In the cross-section, Fig. 2, SW are the slip-ways, and the

Fig. 2.



structure above them, giving temporary support to the ship, is the cradle.

90. The *Slip-ways*, SW, SW, Fig. 2 (also called *sliding-ways*), are a pair of parallel inclined platforms of timber, firmly founded on the floor of the slip, and kept steady in their positions by shores, marked SH. Their slope ranges from 1 in 12 for the smallest ships, to 1 in 24 for the largest. The planks which form the upper surfaces of the slip-ways should have their butt joints bevelled so as to lean a little forward, in order to prevent obstruction in the event of the sinking of the plank of the slide at the fore side of a butt. The ordinary breadth of each slip-way for large vessels is from 3 to 4 feet. The best method, however, of adjusting their breadth, is the following—the area of bearing surface of the *bilge-ways*, or lowest pieces of the cradle, upon the slip-ways should be such that the mean intensity of the pressure shall not exceed 50 lbs. on the

square inch: giving about 45 square inches of bearing surface for each ton of load. This is necessary in order that the pressure of the load may not force out the lubricating material from between the slip-ways and the bilge-ways. The usual distance of the slip-ways from each other, from centre to centre, is about one-third of the extreme breadth of the ship; but this may be varied according to circumstances.

Upon the slip-ways are bolted the *ribands*, R, R (Figs. 2 and 3), so as to form two ledges to guide the cradle during the launch. Shores, marked SH, abut against the ribands outside, to keep them in their places.

When a ship is to be launched in a direction at right angles to a quay or to the water's edge, the lower ends of the slip-ways lie in one straight line perpendicular to the keel of the vessel. But when the vessel is to be launched obliquely to the water's edge, it is often convenient to make the lower ends of the slip-ways lie in a line parallel to the edge of the water, as A B in Fig. 1: care being taken that the upper ends of the bilge-ways lie in a line parallel to the lower ends of the slip-ways, so that both bilge-ways may quit their bearing on the slip-ways at the same instant. (This modification of oblique launching is due to Mr. J. R. Napier.)

The lower ends of the slip-ways usually run into a depth of water such, that by the time the bilge-ways quit their bearing on the slip-ways, the ship shall be completely afloat, with her fore-foot clear of the ground. But if, in order to launch the ship over a quay, or to save expense by using a short cradle, the bilge-ways are so designed as to quit their bearing before the ship is completely afloat, a block must be placed under water in the prolongation of the centre line of the slip, to support the forward part of the keel and the fore-foot during the latter part of the process of launching, after the bilge-ways have quitted their bearing on the slip-ways. That block is usually made of cast-iron, covered with a malleable-iron plate on its upper surface, to protect it against being cut into by the keel of an iron ship.

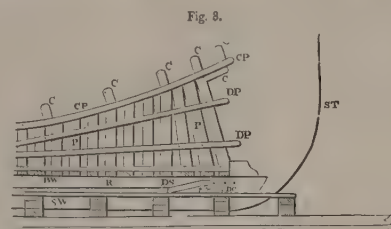
91. *Cradle*.—The base upon which the whole cradle stands consists of the two bilge-ways, B W, B W. (As to the position of their upper ends for oblique launching, see the preceding Article.) The length of the bilge-ways for wooden ships is usually about $\frac{2}{3}$ of that of the ship: for iron ships, about $\frac{1}{3}$ of the ship's length is enough. Their extent of bearing surface has been stated in the preceding Article.

At the middle portion of the ship's length, where her floor is comparatively flat, she is supported upon the bilge-ways by the aid of pieces, called *stopping-up pieces*, laid horizontally, and formed at their upper sides to fit the bottom of the ship.

Before and abaft the stopping-up pieces, the ship is supported on the bilge-ways by means of upright or slightly raking square posts called *poppets* (marked P in Fig. 2). The lower ends of the poppets are kept in their places by being tenoned into the *sole-pieces*, S P. The upper ends of the poppets abut on the bottom of the ship, and are prevented from slipping upwards by the planks, C P; and these are kept steady by means of the *cleats*, C, screwed or bolted in a temporary way to the bottom of the ship. They may also be braced together athwartships by chains passing under the keel. The poppets are braced longitudinally by means of *dagger planks*, marked D P in Fig. 2. When a short cradle is used, as already mentioned, poppets are seldom required.

S are the wedges, called *slices*, between the sole-pieces and the bilge-ways, by means of which the weight of the ship is lifted off the keel, K, and building-blocks, and caused to rest on the cradle alone. When the building-blocks are capable of being easily lowered, the slices are unnecessary.

Fig. 3 is a side view of a small part of the cradle at and near the ship's fore-foot. In addition to the parts lettered, as



already described, ST is the stem of the ship; DS is one of the two *dog-shores*; its lower end abuts against the upper end of the riband of the sliding-way, and its upper end against the *dog-cleat*, D C, which is bolted to the side of the bilge-way. Below the dog-shore is a small block called the *trigger*, which prevents the dog-shore from falling till the moment has arrived for launching the ship. Care should be taken to give the dog-shores no more slope than is just necessary to enable the dog-cleats to clear the ribands of the sliding-ways; lest the dog-shores should *trip*, or turn over backwards, and the vessel be launched unawares. From $\frac{3}{4}$ inch to 1 inch of freedom is left at each side between the bilge-ways and the ribands, increasing gradually downwards.

92. *Preparations for Launching*.—The slip-ways are completed, and the cradles built, and fitted up in their places, a day or two before the launch is to take place: the ship still resting on the building-blocks, and kept upright by props and shores. Shortly before the launch, the bilge-ways are *turned out* of their places on the slip-ways, and the slip-ways and bilge-ways are payed over first with hard tallow and then with soft soap: the tallow to stop the pores of the wood and give it a smooth surface, and the soft soap to lubricate that surface. The bilge-ways are then *turned in* again, and the cradle fitted up as before; and the exposed parts of the slip-ways are covered with boards, to protect them from dirt until the time for launching arrives. If the cradle has slices, they are then driven, so as to *set up* the cradle against the bottom of the ship, and make it support her weight: if not, the same purpose is effected by lowering the building-blocks. Then the props and shores which kept her steady when building, are removed; and the building-blocks are struck one by one from under the keel, commencing at the stern of the ship. Sometimes a few blocks are left under her fore-foot, to be tripped or overturned when she is launched. Meanwhile, if she is to be launched into narrow waters, her bower-anchors have been fixed in the ground of the building-yard, as shown in Fig. 1, and if necessary loaded, to increase their hold; and her chain-cables have been ranged alongside of her, so that their friction on the ground and the hold of the anchors may gradually stop her way after she has been launched. The parts of the two cables nearest the ship's sides should be triced up to prevent their getting foul of the cradle, by means of small ropes which give way one by one as the ship takes the water. If she is to be launched into a wide

space, her anchors are hung at her bows, and are not let go till after she is afloat. Her whole weight now rests on the bilge-ways, and she is kept at rest solely by the dog-shores. By careful inspection it is ascertained that all is clear for launching.

93. *Launching*.—The triggers are removed; the dog-shores

are struck down; and the ship is named as she glides into the water.

Should the ship at first refuse to move, she may be pushed off by the aid of hydraulic presses. After the launch, the cradle floats up from below her bottom, and is hauled ashore by means of ropes attached to it for that purpose.

CHAPTER VII.

SHIPBUILDING YARDS, DOCKS, ETC.

94. *General Arrangement of Yard*.—In a shipbuilding yard where ships are to be built on slips upon the surface of the ground, it is desirable that all that part of the ground which may at any time have to be used for building-slips, should slope uniformly towards the water at an inclination equal, or nearly equal, to the steepest intended inclination of the sliding-ways. (See Article 57 of this Division.)

In order that the errors of the compass in iron vessels may be as small as possible, building-slips for them ought, when practicable, to lie in the magnetic meridian with the stern of the ship towards the nearest pole of the earth; and every iron-clad ship should be armour-plated with her head in the contrary direction to that in which it lay while building.

When there is a quay, embankment, or roadway between the yard and the water, small and light vessels may sometimes be launched over it; but for the purpose of launching large and heavy vessels, it is necessary to make a temporary opening in the quay or bank. Where there is no stone or timber quay, but only an earthen bank, with or without a dry-stone facing, such an opening may be made when required simply by digging, and closed after the launch has taken place by reconstructing the bank and restoring it to its former condition; but where there is a quay, there should be a passage through it, closed by means of a timber or iron gate, with a bridge or platform for a roadway on the top, to be wheeled or floated out of the way when a vessel is to be launched.

A building-slip is sometimes covered with a shed supported on tall posts, to shelter the ship in progress and the workmen from the weather. This is specially useful where a wooden ship is to be left in frame for a time to season before putting on the planking; for long-continued exposure to the alternate action of the sun and rain would be injurious to the framing of the ship.

The designing and erection of such building-sheds is a matter of civil engineering. Their framework may often be made use of to support travelling cranes, for carrying the pieces of material to their proper places in the ship, and also to support scaffolding for the use of the workmen.

In building large ships, much of the cost of scaffolding and gangways round the ship may be saved by using moveable platforms, hanging by tackles between properly stayed poles; each such platform is hauled up or down to the place where it is wanted for the time.^o

The remainder of the yard, where workshops and stores are to stand, should be level, or nearly so. In laying out the various parts of the yard, one object aimed at should be, to make every piece of material, as far as possible, travel continuously onward from its entrance into the yard to the ship of which it is to form part; hence, in general, the entrance should be at the end of the yard furthest from the water: next the entrance should be the stores for raw material; then the workshops and tool-sheds, in the order of the operations performed in them; and then the ground for building-slips. Mould-lofts, pattern-rooms, model-rooms, drawing-offices, and stores for tools, and for finished parts of the ships' equipments, should be placed so as to be easily accessible both from the workshops and from the building-slips, yet so as not to interrupt the direct communication between them.

When a number of machine-tools can be kept constantly at work, upon a succession of pieces of material that are carried continuously onward from the store to the tool and from the tool to the ship, those tools are most economically driven by means of one steam-engine, the power from which is transmitted to the several tools through shafting and belts. But when such tools are employed only at intervals, and upon pieces of material that are carried back and forward between the ship and the tool, a great saving of time and labour is effected by the use of portable tools, capable of being set up near the spot where their work is required; each driven by a small steam-engine forming part of the portable machine, and supplied with steam through underground steam-pipes, properly protected against loss of heat. By laying such pipes with bends in them at intervals, the inconvenient effects of expansion are avoided. On this subject, as well as that of machine-tools in general, reference may be made to a paper by Mr. James Fletcher, of Manchester, in the "Proceedings of the Institution of Mechanical Engineers," 3rd August, 1864.

95. *Building or Graving Docks*.—The design and construction of docks for the building and repair of ships are matters of civil engineering; but some general principles regarding them may nevertheless be stated in this Treatise.

An ordinary dock for shipbuilding is usually a chamber with a floor and walls of stone-masonry, having an opening towards the adjoining harbour, which can be closed when required by means of a pair of folding gates, opening outwards, or by a floating caisson gate. The floor is sometimes made of cast-iron (for an example of which, see a paper by Mr. Milne

^o This method appears to have been first used by Mr. Scott Russell in building the *Great Eastern*.

in the "Transactions of the Institution of Engineers in Scotland" for 1859-60). The floor is nearly as broad as the broadest ship which it is intended to build or to repair in the dock, and at least as long as the longest ship; sometimes a dock is made long enough to hold two or more vessels at the same time. The sides of the dock rise in a series of steps at an angle of about 45°, for the convenience of getting access to it, and of obtaining abutment for shores and support for scaffolding; so that the width of the dock at the top of its side-walls is usually about equal to the width of the floor added to twice the depth of the dock. This, however, does not apply to the entrance or gateway, which has vertical sides. The upper surface of the floor has usually a slight rise towards the middle, in order to make water run off it into a pair of drains at its sides, and often a slight declivity from head to stern.

The dock is filled when required, by admitting water into it through pipes or culverts at the time of high water; and then vessels can be floated in or out. It is emptied partly through the pipes or culverts by the fall of the tide, and partly by pumping. To dry the dock expeditiously, the water is run off into a reservoir, the level of which is below that of the dock floor. The water from the reservoir may be pumped out at leisure. Several docks may be drained into one reservoir.

In a dry-dock, the ship is supported, whether for building or for repairs, by building-blocks, which differ from those used in a slip merely in being almost on a level, instead of forming a steep slope, and by props and shores, abutting against the steps of the dock-walls, and against the bottom, bilges, and sides of the ship. When a ship built in this manner is complete, the water is let into the dock until it is full; and at the time of high-water, the gates are opened and the ship floated out.

A building-dock is often covered with a shed.

When a ship is to be repaired in a dock, the blocks are first laid along the centre-line of the floor, so as to be ready to receive her, and are lashed or ballasted down to prevent them from floating up. The water is then let in; and when the dock is full, the gates are opened, the ship floated in, and the gates closed again. The ship is very carefully guided by means of guy-ropes so as to lie exactly in the centre of the dock, and is also trimmed so as to float exactly upright. A tier of shores called *breast-shores* are then put in, sloping very slightly upwards towards the ship, and abutting at their outer ends or *heels* against the sides of the dock, and at their inner ends or *heads* against the sides of the ship, on a level with that tier of deck-beams which is next above the surface of the water. The water is then slowly let out of the dock, so as to let the ship settle down gently until she bears upon the blocks; when wedges are driven behind the heels of the breast-shores, so as to set them or make them bear firmly against the sides of the ship, care being taken still to keep her upright. As the water continues to fall, additional tiers of shores called *diagonal-shores* and *bilge-shores* are put in; each tier of shores being under the intervals between the shores of the tier next above. In shoring an iron ship care must be taken to place the heads of the shores against the frames, keelsons, or bulkheads, to prevent the indentation of the thin plates.

Each of the blocks can be removed and replaced when required, by the aid of the templates or wedges already described in Article 57 of this Division. *Hydraulic blocks*,

however, being small hydraulic lifting-presses, are more easily adjusted; and they can also be used for straightening a ship which has become hogged or sagged.

96. A *slip-dock* is intermediate between a dock and an open slip; being a dock whose floor slopes towards the water, so that its lower end is in deep water, and its upper end at or above high-water mark. It is more convenient for building and launching, and especially for repairs, than an open slip, and less expensive than a dry-dock. The rate of inclination, where there is sufficient space, may be uniform, so that the floor of the slip is an inclined plane; but in the event of there not being room enough for that construction, the longitudinal section of the floor may be formed to a concave arc of a large circle—a modification introduced by Mr. R. B. Bell. The gates may be either at the lower end or further up, according to convenience. The cradle is a strongly-framed carriage of timber or iron; the main pieces of its frame are a central longitudinal beam, or *keel-beam*, and two parallel *bilge-beams*, corresponding to the bilge-ways of an ordinary cradle. Those pieces are properly connected together by cross-beams and diagonal braces, and are supported by four parallel rows of small wheels, running on four parallel rails, which rest on the floor of the slip. The two middle rails are near each other, and have between them a ratchet, into which several pauls drop from the bottom of the cradle, to prevent it from running back while it is being hauled up. The ship is supported on the cradle by blocks, poppets, and wedges, as on the cradle of an ordinary launch. The cradle is hauled up the slip by means of a chain, which, in the slip first invented by Messrs. Morton, was worked by a wheel-work purchase. Mr. Daniel Miller first introduced the improvement of hauling it by means of a hydraulic press. In the *hydraulic slip* (as this is called), the cylinder of the press lies resting on a strong foundation at the upper end of the slip-way: its plunger has a cross-head, connected by means of side-rods with a cross-tail. To the cross-tail is attached the chain for hauling up the cradle, consisting of a series of long straight links or traction-rods, each equal in length to the stroke of the plunger, and connected together by pins. The plunger is driven by water forced in by a set of pumps worked by a steam-engine. After a forward stroke of the plunger has been completed, an escape-valve is opened for the water, and the plunger is drawn back by a counterpoise; one length of traction-rod is disconnected and taken out; and when the plunger has completed its return stroke, the shortened chain of traction-rods is again connected with the cross-tail, to be ready for the next forward stroke; and so on. The operation of disconnecting the traction-rods, taking out a length, and re-connecting them, is called *fleeting*. The usual speed of hauling up large ships, including stoppages for fleeting, is about four feet per minute. The tractive force required may be estimated by multiplying the whole load hauled up by the steepest rate of inclination of the slip, and adding $\frac{1}{10}$ of the same load for friction. (For details, see a paper by Mr. R. B. Bell, in the "Transactions of the Institution of Engineers in Scotland" for 1858-59.)

97. *Floating Docks, Saucers, &c.*—A floating dock is usually an iron vessel of a rectangular shape: the bow being somewhat rounded or pointed, to diminish the resistance of the water to its motion. At the stern end is a strong caisson gate. The vessel has a double skin: the inner skin forms a basin or dock large

enough to contain and float the ships which are to be repaired in it; the outer skin is larger still, so as to inclose a space between the two skins sufficient to give the required buoyancy to the dock, in order that it may be floated from place to place with a ship within it, or sent under steam or sail from the place where it is made to the place where it is to be used. In a well-proportioned floating dock, the outer shell is one-third broader and one-third deeper than the inner. The space between the shells is subdivided by iron bulkheads; and the skins and bulkheads are proportioned and arranged so as to give the required strength, according to the principles of the Third Division of this Treatise. The weight, or unloaded displacement, of an iron floating-dock, designed with due regard to strength and economy of material, may be estimated at from $\frac{1}{10}$ to $\frac{1}{15}$ of its load displacement, leaving from $\frac{5}{10}$ to $\frac{4}{10}$ for lading.

A floating dock is grounded and floated when required, by letting water into the space between its two shells, and by pumping that water out again by steam-power.

A *saucer* or *camel* is like the bottom of a floating dock, without sides, or with low sides; so that when it is grounded in sufficiently deep water, a ship can be floated over it without the necessity of opening and shutting gates. A method of lifting the saucer and setting it afloat with the ship on it, invented by Mr. Edwin Clark, consists in raising it from the bottom, by means of two rows of vertical hydraulic presses, to such a height that the water runs out of it. The escape-valves are then closed, and the saucer, with the ship on it, is floated away. One set of hydraulic lifting presses with their steam-pumps will thus successively lift and set afloat any number of saucers with ships on them.

ADDENDA TO THE FIRST DIVISION.

CHAPTER II, SECTION I. (MENSURATION OF AREAS AND VOLUMES, ETC.)

I. CORRECTED TRAPEZOIDAL RULE.—To measure a plane area in separate divisions, standing on equal intervals of the base-line:

Divide the length of the base-line into any number of equal intervals; measure the ordinates, or breadths perpendicular to the base-line, at its ends, and at the points of division, and set them down in a table in their order.

Take the *first differences* of the successive breadths, marking them as positive (+) or negative (−) according as the breadths are increasing or diminishing, and writing the difference of each pair of breadths opposite the interval between them.

Take the *second differences*, or differences of the successive first differences, marking them as positive (+) for increasing positive and diminishing negative first differences, and as negative (−) for increasing negative and diminishing positive first differences; and writing each second difference opposite to the interval between the pair of first differences to which it belongs.

Each intermediate breadth will now have a second difference opposite to it. Fill up the blanks opposite the two endmost breadths, with second differences, each in arithmetical progression with the two second differences next it, if these are unequal, or equal to them, if they are equal.

Divide each second difference by 12, and write the quotients in a column headed *corrections*; marking each correction with the *reverse* sign to that of the second difference of which it is the $\frac{1}{12}$ part.

Add each correction to, or subtract it from, the breadth opposite it, according as the correction is marked + or −. The results may be called *corrected breadths*.

To find the area of the division of the figure contained between any two ordinates: multiply the *half sum* of the *corrected breadths* by the *interval between them*.

For the area of the whole figure, add together all the divisions.

This rule is exact for parabolic figures of the second and third order, and approximate for other figures.

By substituting *sectional areas* for *breadths*, the volume of a solid figure (such as the displacement of a ship) may be calculated in separate layers. (See a paper by C. W. Merrifield, Esq., F.R.S., in the "Transactions of the Institution of Naval Architects for 1865.")

EXAMPLE.—Length of base 120 feet, divided into six intervals of 20 feet:—

Breadths. Feet.	First Differences.	Second Differences.	Corrections.	Corrected Breadths.	Area. Sq. Feet.
17.28	...	(− 1.92)	...	17.44	...
...	− 0.88	...	+ 0.16
16.40	...	− 1.44	+ 0.12	16.52	339.6
...	− 2.32	...	+ 0.08
14.08	...	− 0.96	+ 0.08	14.16	306.8
...	− 3.28	...	+ 0.04
10.80	...	− 0.48	+ 0.04	10.84	250.0
...	− 3.76
7.04	...	0	0	7.04	178.8
...	− 3.76
3.28	...	+ 0.48	− 0.04	3.24	102.8
...	− 3.28
0	...	(+ 0.96)	− 0.08	− 0.08	31.6
Total area,.....					1209.6

The numbers in parentheses at the top and bottom of the column of second

differences, are those filled in by taking each of them in arithmetical progression with the two adjoining second differences.

The last corrected breadth being *negative*, is subtracted instead of added in computing the area of the last division. This often happens where the actual breadth is small, or nothing, and the outline of the figure concave.

II. WOOLLEY'S RULE FOR VOLUMES.—To measure the volume of a solid figure standing on a plane base:

Divide the base into equal and similar rectangles; measure the thickness at the centre of each rectangle, and at the middle of each of its four sides.

Then to calculate separately the volume of any rectangular subdivision of the solid: add together the thicknesses at the middles of the four sides, and twice the thickness at the centre; divide the sum by six, and multiply the quotient by the area of the rectangular subdivision of the base.

To calculate the whole volume: add together the volumes of the sub-divisions:—or otherwise, add together all the thicknesses at the outside boundary of the base, and the doubles of all the other thicknesses; divide the sum by six, and multiply the quotient by the area of one rectangular subdivision of the base.

This rule is exact where the thickness of the solid is a rational integral function of co-ordinates upon the plane base, not exceeding the third degree, and approximate for other figures.

III. MERRIFIELD'S RULE FOR DENSITIES.—The mean density of a rectangular solid is $\frac{1}{6}$ part of the sum of the six densities at the centres of its six faces.

This rule is to be used in computing the weight and moment of any body whose density varies continuously. It is exact when the density is a rational integral function of the co-ordinates not exceeding the third degree, and approximate for other functions. (See "Transactions of the Institution of Naval Architects for 1865.")

IV. RULE FOR MEASURING THE LENGTHS OF CURVED LINES.—Divide by the eye the curved line into any *even* number of nearly equal intervals. Measure the total length of the series of straight chords joining the ends and the alternate points of division. Measure also the total length of the series of straight chords joining the ends and all the points of division (which length will be greater than the former). To the latter, or greater length, add *one-third* of its excess above the former; the sum will be the approximate length of the curved line.

For a circular arc, divided into intervals of 30°, the error of this method is about $\frac{1}{1000}$; and it varies nearly as the fourth power of the angular interval.

V. RULE FOR SETTING-OFF A GIVEN LENGTH FROM A GIVEN POINT ON A CURVED LINE.—Take any convenient aliquot part of the given length in the compasses, step the length along the curve from the given point, and mark the point arrived at.

Take one-half of the same aliquot part in the compasses, step the length a second time, and mark the point arrived at, which will be behind the first point.

Take one-third of the distance between the first and second points arrived at, and set it off *backwards* from the second point; this will give a close approximation to the point required.

The error is the same with that of the preceding rule.

The present rule may be used also to measure the lengths of curved lines, by setting-off the nearest whole number of any convenient unit of length which the line will contain, and taking the fractional remainder as if it were straight.

ADDENDA TO THE FOURTH DIVISION.

ARTICLE 69.

A "JURY OR TEMPORARY RUDDER" is sometimes required to replace the original rudder in the event of its being lost at sea. The following description of a method of rigging one, put in practice by Mr. Christopher Pottinger, is extracted from the *Practical Mechanic's Journal* for 1859.

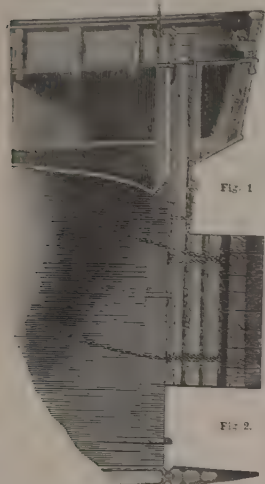


Fig. 1 is a side elevation, and Fig. 2 a plan of the Jury-rudder:—

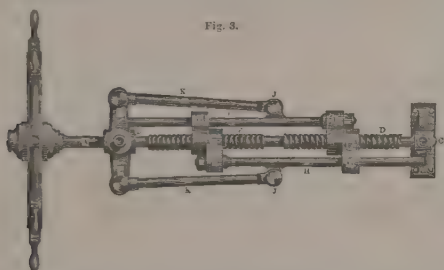
The main piece was made out of a length of about 27 feet of a spare topmast; two lengths of 12 feet each of the same spar were bolted thereto, and planked over with rough planking to complete the rudder. It was then hung to the stern-post by two gudgeons, made out of the kedge anchor stock, fitted in such a manner that the shoulder of the anchor stock bore the weight, and the forelock served as a woodlock for the rudder. The stream chain was used as the third gudgeon and brace, the brace having been carried off the stern-post; it had a round turn around the rudder stock, crossed over the stern-post, and was set up to the fore-chains by runner and tackle. The topsail-sheet chain was fixed round the rudder with a clove hitch on the after part, to be used as a pendant to steady the rudder in case it should be found necessary to heave-to the vessel in stress of weather.

When finished, this rudder resembled the rudder of a flat or canal boat; it was broader than the original rudder of the vessel, but did not go so far down; and was found to steer the ship as well as the original rudder, with the same power at the wheel.

ARTICLE 70.

Screw Steering Gear.—The following figures represent two forms of apparatus for steering, by means of right-handed and left-handed screws.

(*Mr. William's Gear.*) In Fig. 3, A C is the shaft or spindle of the steering-wheel; D, right-handed screw; E, left-handed screw; F, F, two traversing nuts;



H, H, two traversing rods, parallel to the spindle, fixed respectively to the two traversing nuts at I, I; G, G, guide-eyes for the opposite ends of the traversing

rods; K, K, two links jointed respectively at J, J, to eyes on the traversing rods, and at L, L, to the two arms of the yoke; B, rudder-head.

(*Reed's Gear.*) In Fig. 4, A is the spindle of the steering-wheel, with two screw-threads cut upon it, one right-handed and the other left-handed. B, B, nut-blocks, having semicircular recesses in their inner faces, in which are cut threads; a left-

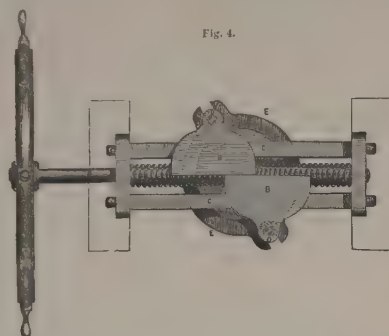


Fig. 4.

handed thread in the starboard nut-block, and a right-handed thread in the port nut-block. The nut-blocks slide on two guide-bars, C, C. Near each nut-block, at D, a pin projects downwards, and works between a pair of lugs projecting from the cap, E E, of the rudder-head.

ARTICLE 77.—ORDINARY WEIGHTS OF SHIPS' BOATS.

CARVEL-BUILT.		Tons and Decimals.
Launch,.....	
Barge,.....	4.0 to 5.5
Pinnace,.....	about 1.5
Yaw,.....	1.25 to 1.5
.....	0.8 to 1.15
CLINKER-BUILT.		Tons and Decimals.
Gig,.....	
Cutter,.....	0.4 to 0.7
Jolly-boat,.....	0.5 to 0.75
.....	0.2 to 0.5

ARTICLE 96.

The *Telescopic Cradle* for slip-docks,* is a cradle on wheels divided into several parts (like a train of wagons), which, by means of coupling-rods sliding in tubes, can be attached together at different distances apart, in order to adapt the cradle to vessels of different lengths. When this cradle is run to the lower end of the slip under the bottom of a vessel, the coupling-rods are pushed home, so as to shorten the cradle as much as possible; and thus a shorter length of slip-ways is required than would otherwise be necessary. As soon as the vessel bears upon the foremost division of the cradle, the heaving-up is begun, and the coupling-rods are drawn out in succession to their full length, as one division after another is set in motion.

* Introduced by Mr. Robert Turnbull.

DIVISION FIFTH.

MASTS, SAILS, AND RIGGING.

CHAPTER I.

GENERAL DIMENSIONS, FIGURE, AND ARRANGEMENT OF MASTS AND SAILS.

SECTION I.—GENERAL PRINCIPLES.

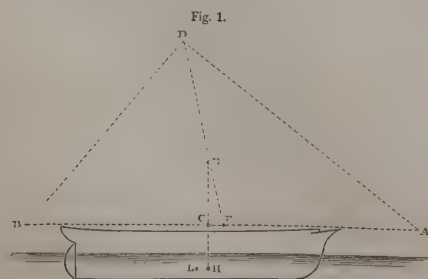
ARTICLE 1. *Equivalent Triangle*.—In Chapter V., Section V., of the First Division, Articles 182 to 185, and in Section VI. of the same Chapter, Article 190, the mode of action of the wind in propelling a vessel by means of sails has been explained. Reference may be made particularly to Article 182, in which are shown the relations between the *moment of sail* and the stability of the ship, and between the *area of sail* and her resistance.

In the present Chapter it has to be considered how to design a set of sails for a given vessel, so that the moment of sail shall be properly adapted to her stability.

As already explained in Article 182 of the First Division, the horizontal dimensions, or breadths, of the several sails of a ship, are regulated chiefly by her length; and the vertical dimensions, or heights, by her breadth, or rather by the height of the *centre of effort* of the sails, which depends mainly on the ship's breadth.

From measurements of the sails of a great many examples of actual vessels rigged in a variety of different ways, it appears that the area and moment of *all plain sail* are, in almost every case, nearly equal to those of what may be called an *equivalent triangle*, constructed as follows:—

Let Fig. 1 represent the longitudinal vertical section of a vessel: L, the centre of the immersed part of that section,



assumed to be at the same level with the *centre of lateral resistance*; E, the *centre of effort* of the sails (see Division First,

Article 181); EH, a vertical line through E, cut by LH, a horizontal line through L, so that EH is the *leverage of sail*.

At the level of the highest point of the plank-sheer, draw a horizontal line, AB, as the *base of sail*.

In a vessel fully equipped with sails as her only or chief propelling power, A may be taken directly below the tack, or foremost corner, of the jib, and B at the clew, or after corner, of the driver, or other aftermost sail.

In a vessel only partially equipped with sails, her chief propelling power being steam, the base of sail may be interrupted; and then the lengths of its several parts are to be measured and added together.

Bisect AB in F; join FE, and produce it to D, making $FD = 3 FE$; join AD, BD. Then E will be the centre of the triangle ABD; and from the measurements before referred to, it appears that the actual area and moment of sail are almost always nearly equal respectively to the area and moment of that triangle; the extent of deviation of the practical examples being about $\frac{1}{15}$ part each way.

Let EC be the height of the centre of effort above the base of sail. This may be called the *mean height of sail*. Then—

- I. $\begin{cases} \text{Area ABD} = AB \times EC \times 1\frac{1}{2}; \\ \text{Area of sail} = AB \times EC \times \text{from } 1\cdot4 \text{ to } 1\cdot6 \\ \text{(average about } 1\cdot5); \end{cases}$
- II. $\begin{cases} \text{Moment of ABD} = AB \times EC \times EH \times 1\frac{1}{2}; \\ \text{Moment of sail} = AB \times EC \times EH \times \text{from } 1\cdot4 \text{ to } 1\cdot6 \\ \text{(average about } 1\cdot5). \end{cases}$

The following are examples of the ordinary proportions borne by the length of the *base of sail* to the *length of the line of flotation*:—

Vessels of the old proportions; the length being between $3\frac{1}{2}$ and 4 times the breadth, viz.:—		AB
		Line of Flotation.
Ships,.....	from 1·5	to 1·6
Brigs,.....	" 1·6	to 1·65
Schooners,.....	" 1·6	to 1·7
Cutters,.....	" 1·7	to 1·9
Ships of lengths between 5 and 6 times the breadth,	" 1·35	to 1·5

The *middle of the base of sail* is found, in a variety of practical examples, to be *afore* the centre of effort by a distance

(CF, Fig. 1), whose ratio to the length of the base of sail ranges from $\frac{1}{4} = \cdot 025$, to $\frac{1}{8} = \cdot 0625$; the usual limits for ships being from $\cdot 025$ to $\cdot 05$, and for brigs, schooners, and cutters, from $\cdot 05$ to $\cdot 0625$.

For the position of the centre of effort relatively to the middle of the line of flotation, see Division First, Article 181.

The distance of the middle of the base of sail afore the middle of the line of flotation ranges from $\cdot 08$ to $\cdot 15$ of the length of the line of flotation, being least in the bluffest vessels, and greatest in the finest.

Of course such rough approximations as those of the preceding rules, marked I. and II., do not supersede the necessity for making an exact calculation of the area and moment of sail after the rigging plan has been drawn. This will be again referred to further on. The chief use of the equivalent triangle is to determine approximately the centre of effort and area of sail suited for a vessel, when no data are given except the *moment of sail* suited to her stability (as found by the rules of the First Division, Article 182); the position and length of her base of sail, AB; and the position, L, of the centre of lateral resistance. The method of proceeding is as follows:—

III. *To find the Centre of Effort.*—Divide the moment of sail by the base, AB, and by $1\frac{1}{2}$. To the quotient add $\frac{1}{2}$ of the square of the height, CH, of the base of sail above the centre of lateral resistance, and extract the square root of the sum. To that root add $\frac{1}{2}$ CH; the sum will be the height, EH, of the centre of effort above the centre of lateral resistance.

The approximate area of sail is then to be found by Rule I.; and the moment may also be computed by Rule II., to check the accuracy of the calculations.

EXAMPLE.

Displacement of ship,.....	2160 tons.
Metacentric height,.....	4 feet.
Angle of heel in circular measure (about 4°),.....	0·07.
Length of base of sail, AB (the line of flotation) being 200 feet,.....	280 feet.
Height, CH, of base of sail above centre of lateral resistance,.....	24 feet.

By the rule of Article 182 of Division First—

$$\text{Moment of sail} = 2160 \times 2240 \times 4 \times \cdot 07 = 1,354,752.$$

Then by Rule III. of this Article—

$$\begin{aligned} \frac{1,354,752}{280 \times 1\frac{1}{2}} &= \dots\dots\dots 3225\cdot6 \\ \text{Add } \frac{CH^2}{4} = 11^2 &= \dots\dots\dots 121\cdot0 \\ \text{Sum} &\dots\dots\dots 3346\cdot6 \end{aligned}$$

The square root of which is 58, nearly. Then—

$$\begin{aligned} 58 + 11 &= 69 = \text{EH, leverage of sail; and} \\ 58 - 11 &= 47 = \text{EC, mean height of sail.} \end{aligned}$$

Then by Rule I. of this Article—

Approximate area of sail = $47 \times 280 \times 1\frac{1}{2} = 19,740$ square feet;
and by Rule II.—

$$\text{Approximate moment of sail} = 19,740 \times 69 = 1,362,060;$$

which nearly agrees with the given moment.

In applying all the preceding rules, it is to be observed that the area of sail in the calculations is exclusive of all parts of sails which are becalmed, or screened from the wind by other

sails; and this must be attended to in every case in which a sail is so shaped that part of it is concealed by another sail in the rigging plan.

2. *Names, Numbers, and General Proportions of Masts.*—Masts are usually one, two, or three in number, and in a few cases only, four and upwards. Those numbers do not include the *bowsprit*, which projects from the bow of the vessel.

A single mast is called simply the *mast*. When there are two masts, and the aftermost is the larger, it is called the *main-mast*, and the other the *foremast*; when the aftermost of two masts is the smaller, it is called the *mizenmast*, and the other sometimes the *mainmast* and sometimes the *foremast*. Three masts are called in their order from the ship's head, the *foremast*, the *mainmast*, and the *mizenmast*. Four masts are sometimes called the *foremast*, the *mainmast*, the *main mizenmast*, and the *bonaventure mizenmast*; but it will be a long time, probably, before fixed names be given to the several masts when their number exceeds three.

From practical examples it appears, that the proportions of the different masts of a vessel to each other seldom deviate much from the following average values:—

Mizen	:	Main	:	Fore
: : 7	:	10	:	9

3. *Divisions of Bowsprits and Masts.*—Some vessels have a bowsprit in one division only. When the length of a bowsprit consists of two or more divisions, the term *bowsprit* is applied, strictly speaking, to the aftermost division only, which is fixed; the next piece is called the *jib-boom*; and when there is a third piece, it is called the *flying jib-boom*: the last two are capable of being run in and out.

A mast whose height consists of one piece only, is called a *pole*, or *pole-mast*; and the term *pole* is applied also to the uppermost part of any mast.

When a mast in height consists of two or more pieces, the lowest piece, which is fixed in the vessel, is called the *lower mast*; the other pieces, which are capable of being raised and lowered, are called in succession the *topmast*, the *topgallantmast*, and the *royal mast*. When a topgallantmast and royal mast are in one piece, the upper part is called the *royal pole*.

The height of a lower mast consists of three subdivisions: the *housing*, from the *heel* of the mast, where it rests on the *step*, to the upper surface of the upper deck, being the part within the ship; the *hounding*,* from the upper deck to the *hounds*, where the rigging is secured to the mast; and the *head*, being the part to which the lower end of the topmast is secured, from the hounds to the *cap*.

The height of a topmast, or of a topgallantmast, consists of two subdivisions: the *hounding*, from the heel to the hounds; and the *head*, from the hounds to the cap.

The length of the head of a mast ranges from $\frac{1}{10}$ to $\frac{1}{8}$ of its length from heel to hounds; but it is really governed by the length of the topmast and spars above.

4. *Stations of Masts.*—The stations for the masts of a vessel are usually defined by stating the proportions which their distances from the middle of the load-water-line, or line of flotation, bear to the length of that line. The best values for those proportions have been fixed within narrow limits, by long experience.

* The term *hounding* is often applied to the total height of the two lower subdivisions, from the heel to the hounds.

They are on the whole greater in broad and full vessels than in long and fine vessels. The following table shows their most common values :—

	Distance from Middle of Line of Flotation in Fractions of Length of that Line.	
	Aft.	Afore.
ONE MAST,	—	{ from 0·10 to 0·14
TWO MASTS; MAIN AND MIZEN :—		
Mainmast,	—	{ from 0·10 to 0·14
Mizenmast,	{ from 0·4 to 0·5	—
TWO MASTS; FORE AND MAIN :—		
Foremast,	—	{ from 0·30 to 0·40
Mainmast,	{ from 0·10 to 0·15	—
THREE MASTS :—		
Foremast,	—	{ from 0·3 to 0·4
Mainmast,	{ from 0·03 to 0·08	—
Mizenmast,	{ from 0·30 to 0·40	—

5. *Rake of Masts—Steeve of Bowsprit.*—The rake of a mast is its inclination to the vertical, forward or aft. The only masts which rake forward, are the foremasts of vessels of what is called the "lateen rig;" and in them the rake ranges from $\frac{1}{2}$ to $\frac{1}{3}$ of the length. In all other cases the rake of masts, if any, is aft, and is commonly greatest in the smallest vessels; it increases from the foremast to the mainmast, and from the mainmast to the mizenmast. Examples of its amount will be given in describing different styles of rig. It has the effect of throwing the centre of effort of the sails somewhat further aft than if the masts were vertical, and stepped in the same place.

The chief use of making the masts rake aft, seems to be to obtain a position more advantageous to the strength of the rigging than the vertical position.

The *steeve* of the bowsprit is the rate at which it slopes upwards. It varies much in different kinds of rig, and examples of it will be given further on. Its uses are, to enable the inboard part of the bowsprit to be securely fastened, and to keep the out-board part clear of the waves.

6. *Classes of Sails.*—All sails are divided into two principal classes—*square-sails* and *fore-and-aft sails*.

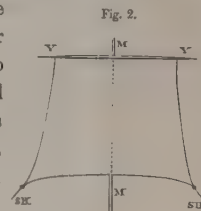
Square-sails are those whose middle position is transverse, or athwartships, and which can be braced to a greater or less angle on either side of that position, to suit the direction of the wind, but cannot be braced exactly fore-and-aft.

Fore-and-aft sails are those whose middle position is longitudinal, or fore-and-aft, and which can be trimmed to a greater or less angle to either side of that position, but not directly athwartships.

Square-sails are the most efficient for running large, or away from the wind, or before the wind; fore-and-aft sails are the best for running near the wind, and for manoeuvring; and for certain manoeuvres they are essential: hence, although some small vessels are wholly without square-sails, none are wholly without fore-and-aft sails. Vessels are said to be *square rigged*, or *fore-and-aft rigged*, according as their principal sails are square, or fore-and-aft.

Every square-sail (with the exception of studding-sails, to be described further on) is of one kind of figure, like that shown

in Fig. 2; that is to say, it is four-sided, and symmetrical at each side of an upright centre line, MM; which line, in the figure, also represents the mast. The upper edge, or *head*, and lower edge, or *foot*, are parallel to each other; the two side edges, called *leeches*, are of equal length. The head is bent to, and hangs from, a transverse spar, YY, called a *yard*. The two lower corners, called *clews*, are stretched by means of ropes called *sheets*, SH, SH (sometimes aided by other ropes called *tacks*, which will be described further on).



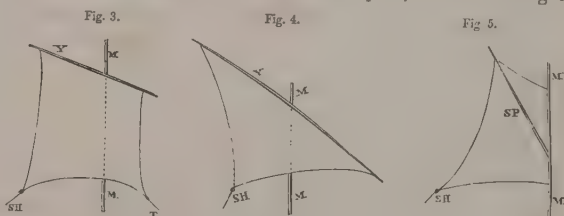
Fore-and-aft sails are of irregular figures, some three-cornered, some four-cornered. Some are bent to yards, some directly to the mast, and some to sloping ropes called *stays*.

The highest corner of a fore-and-aft sail always points aft, and in four-cornered sails is called the *peak* of the sail. The word *peak* is also applied to the end of the yard, sprit, or gaff, to which the peak of the sail is bent. The upper forward corner is called the *throat*, or *nock*. The lower after corner is called the *clew*; the lower forward corner the *tack*; the clew is hauled aft by a rope called the *sheet* (SH, Figs. 3 to 9); and the tack hauled forward by a rope, also called the *tack* (T, Figs. 3 to 9).

In a fore-and-aft sail, the *head* is the upper edge, if the sail is four-cornered, or the upper corner, if the sail is three-cornered; the *foot* is the lower edge; the foremost edge is called the *luff*, or *weather-leech*; the aftermost edge is called the *lee-leech*, or simply the *leech*.

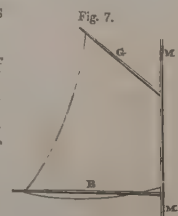
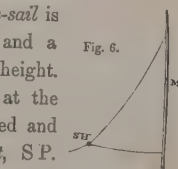
Figs. 3, 4, 5, 6, and 7 are examples of fore-and-aft sails which hang from spars. MM in each figure is the mast.

Fig. 3 is a *lug-sail*, hanging from a yard, which is slung at



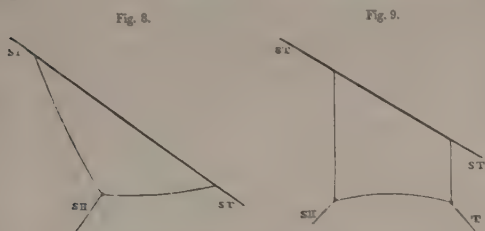
about two-thirds of its length from the peak. A *gaff-topsail* may be regarded as a lug-sail with a narrow head. Fig. 4 is a *lateen-sail*, bent to a *lateen-yard*. A *settee-sail* is intermediate in shape between a lug-sail and a lateen-sail, having a weather-leech of small height. Fig. 5 is a *sprit-sail*, bent to the mast at the weather-leech, and having the peak stretched and held up by a raking spar called a *sprit*, SP. Fig. 6 is a *shoulder-of-mutton sail*, being a triangular sail with the weather-leech bent to the mast. The mast, to carry this sort of sail, is sometimes lengthened by means of a moveable piece, called a *sliding-gunter*.

Fig. 7 is one of the most useful forms of fore-and-aft sails, being a four-cornered sail bent at the weather-leech to the mast, and at the head to the *gaff*, G. The foot of the sail is stretched sometimes by a *boom*, B, and sometimes simply by a *sheet*. Sails of this form have different names, according to their situation and use; such as *gaff-mainsail*, *gaff-foresail*, *spanker* or *driver*,



spencer, trysail, of which the special application will be afterwards shown.

Figs. 8 and 9 are examples of sails bent to stay-ropes; when



three-cornered, as in Fig. 8, the sail is called a *jib* or a *stay-sail*, according to its position. Fig. 9 is a four-cornered stay-sail.

Booms, such as B, are sometimes used to stretch the foot of sprit-sails and lug-sails.

Fore-and-aft sails bent to yards which hang across the mast, such as lug-sails and lateen-sails, are suited to boats and small vessels only; because, when the vessel goes about, it is necessary to shift the sail to the lee side of the mast; which with large and heavy sails is a troublesome and difficult operation.

Sprit-sails, gaff-sails, stay-sails, and jibs, are free from this defect.

The weather-leech of a fore-and-aft sail which is bent to a mast or to a stay, usually hangs to a series of *hoops* or *hanks* which run on the mast or stay, to enable the sail to be hoisted and lowered. An upright spar close abaft of, and parallel to the mast, for such hoops to run upon, is called a *trysail mast*. Gaff-sails sometimes hang from hoops which run on the gaff.

7. *Studding-sails—Bonnet—Ringtail*.—These are names for additional sails spread by the aid of light booms and yards beyond the edges of the principal sails, to increase the area of canvas in light winds. All of them are four-sided. *Studding-sails* are added at one or both leeches of a square-sail; a *bonnet* is added below the foot of the principal sail, to which it is laced; the *ringtail* is spread at the lee-leech of the driver, or aftermost fore-and-aft sail. Studding-sails are not used on a mizenmast; because the braces of the yards on that mast are led forward, and give no assistance in resisting the forward pressure of the wind.

8. *Order and Proportions of Square-sails on a Mast*.—The square-sails spread on the successive divisions of a mast have names corresponding, with a few exceptions, to the names of those divisions. On a lower mast a square-sail is called a *course*; and on the higher divisions, successively a *top-sail*, a *topgallant-sail*, and a *royal*. The royals are the highest sails commonly spread; but in very light airs small square-sails are sometimes hoisted above them, called *sky-sails*, or *sky-scrapers*, or *flying kites*. The yards to which those sails are bent have corresponding names.

The proportions amongst the dimensions of the several square sails on a mast, seldom deviate far from a set of average values, which may be estimated as follows:—

	Ratio of Depth to the depth of the Top-sail.	Ratio of Breadth to the breadth of the foot of the Top-sail.
Royal,	0.4	head } 0.40 foot } 0.55
Topgallant-sail,	0.6	head } 0.75 foot } 1.00
Top-sail,	1.0	head } foot }
Course,		head }

Sometimes a square top-sail is divided into two parts of equal depth, called the *upper* and *lower top-sail* respectively, and bent to upper and lower top-sail yards.

9. *Reefing* consists in taking in part of a sail, so as to reduce its area, and is applied to courses and other lower sails, whether square or fore-and-aft, and to square top-sails. The means of effecting it will be described in a later chapter. Square-sails are usually reefed at the head; fore-and-aft sails at the foot. When a top-sail is *close-reefed*, its depth is reduced to about one-half.

In some cases, to be afterwards specified, instead of reefing a sail, a series of similar sails of different sizes are kept ready for bending, to suit different states of the weather.

10. *Finding Centres and Areas in Detail*.—When the *rigging plan* or *sail-draught* of an intended vessel has been so far prepared as to show her bowsprit and masts in their several stations, and the outlines of her principal sails, the *area* of each sail is to be found by one or other of the following rules:—

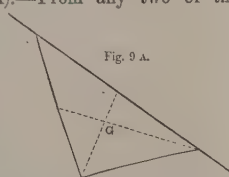
I. **SQUARE-SAILS**.—Multiply the depth by the half sum of the breadths at the head and foot.

II. **TRIANGULAR FORE-AND-AFT SAILS**.—Multiply any side by half its perpendicular distance from the opposite corner.

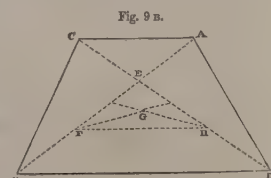
III. **FOUR-SIDED FORE-AND-AFT SAILS**.—Multiply either diagonal by the half sum of its perpendicular distances from the opposite corners.

The *centre* of each sail is then to be found by one or other of the following rules:—

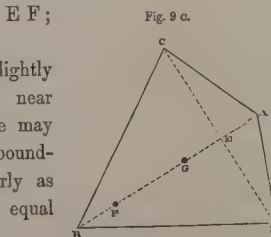
IV. **TRIANGULAR SAILS** (Fig. 9 A).—From any two of the corners draw straight lines to the centres of the opposite sides; the intersection, G, of those lines will be the centre of the sail: or otherwise, from any corner draw a straight line to the middle of the opposite side, and cut off one-third from that line, beginning at the side.



V. **FOUR-SIDED SAILS—Case First** (Fig. 9 B).—Draw the diagonals, AB and CD, cutting each other in E; make BF = AE, and DH = CE; then by Rule IV., find the centre, G, of the triangle, EFH, which will be the centre required.



VI. **FOUR-SIDED SAILS—Case Second** (Fig. 9 C).—When E happens to bisect one of the diagonals, as CD, so that Rule V. fails, make BF in the other diagonal = AE, and EG = $\frac{1}{3}$ EF; G will be the centre required.



When a sail is bounded by slightly curved lines, an approximation near enough for the present purpose may be made by drawing straight boundaries, so as to inclose, as nearly as can be judged by the eye, an equal area having the same centre.

The *centre of effort*, or common centre of all the sails, is then to be found agreeably to the principles of Article 181; that is to say:—

VII. Multiply the area of each sail by the height of its centre above any convenient horizontal line (such as the line of

flotation): divide the sum of the products by the total area of sail; the quotient will be the height of the centre of effort above the same horizontal line:—

VIII. Multiply the area of each sail by the horizontal distance of its centre from any convenient vertical line (such as a line drawn vertically through the middle of the line of flotation); distinguish the products into forward and after, according as the centres of the sails lie before or abaft the vertical line; take the sums of the forward and after products separately, and the difference of those sums; divide that difference by the total area of sail; the quotient will be the horizontal distance of the centre of effort from the vertical line, and will lie before or abaft that line, according as the forward or after products are in all the greater.

IX. The *moment of sail* is then to be computed by multiplying the area of sail by the height of the centre of effort above the centre of the immersed longitudinal section; and by comparing the area and moment as now calculated in detail with the previous estimate of those quantities obtained as described in Article 1, it is to be ascertained whether the design for the sails on the sail draught requires modification in order to adapt it properly to the stability of the vessel.

X. The sails are then to be distinguished into *head-sail* and *after-sail*, and the *centre of head-sail* and *centre of after-sail* are to be found separately, in the same manner with the centre of effort: the proportion which the horizontal distance between those two centres bears to the length of the line of flotation being, as explained in Division First, Article 190, a measure of the handiness of the vessel when manœuvring under sail, and ranging from 0.6 to 0.7 in good examples.

The *areas* of head-sail and after-sail are of course to each other inversely as the distances of their respective centres from the centre of effort. Their relative proportion varies very much in the smaller classes of vessels; in ships, however, it is more nearly uniform; the area of after-sail being greater than the area of head-sail in a ratio which ranges from 3 : 2 to 5 : 3. The greater area of after-sail is advantageous, as counteracting the tendency to check the ship's headway, which is produced when the head-sail, or part of it, is taken aback during the operation of *tacking*, or going about with her head to windward.

10 A.—*Geometrical Construction of Sails*.—The following rules may sometimes be useful:—

I. **SQUARE-SAIL** (Fig. 9 D).—*Given, the foot, BB, the depth, AC, and the area of a square-sail: to construct its figure.* Divide

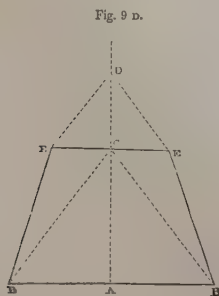


Fig. 9 D.

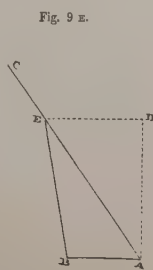


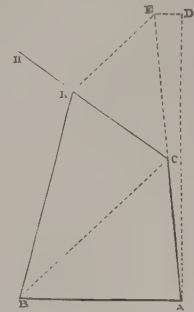
Fig. 9 E.

the area by half the breadth at foot; lay off the quotient, AD, upwards from the foot on the upright centre line. Join CB, CB; and parallel to those lines, draw DE, DE. Through C draw a straight line parallel to BB, cutting the two oblique

lines from D in E, E; join EB, EB; then EE will be the head of the sail, and EB, EB, its leeches.

II. **TRIANGULAR SAIL** (Fig. 9 E).—*Given, the foot, AB, the direction of one leech, AC, and the area of a triangular sail: to construct its figure.* Divide the area by half the foot, and set up the quotient as a perpendicular, AD, to AB. Through D draw DE parallel to AB, cutting AC in E; join EB: ABE will be the required figure.

Fig. 9 F.



III. **QUADRANGULAR FORE-AND-AFT SAIL** (Fig. 9 F).—*Given, the foot, AB, the weather-leech, AC, the direction, CH, of the head, and the area of a quadrangular sail: to construct its figure.* Divide the area by half the foot, and set up the quotient as a perpendicular, AD, to AB. Through D draw DE, cutting AC produced in E. Join CB; and parallel to it, through E, draw EK, cutting CH in K. Join KB. Then ABKC will be the required figure.

SECTION II.—DIFFERENT STYLES OF RIG.

11. *General Remarks*.—The main distinction between square rigged and fore-and-aft rigged vessels has been explained in Article 6 of this Division. The other differences amongst styles of rig depend chiefly on the number of parts into which the whole area of sail is divided.

If efficiency in propelling the vessel were the only thing to be considered, the best style of rig would be that containing the fewest and the largest sails. But the difficulty of working very large sails and spars renders it necessary to subdivide the area of sail into parts, which are the more numerous, the larger the vessel and the fewer the crew. Large ships have always numerous sails; and amongst the smaller classes of vessels, it is often necessary to rig those intended for trading purposes, with smaller and more numerous sails than war-vessels of the same size, because of the former having fewer men to work them.

The simplest styles of rig are fore-and-aft, because fore-and-aft sails are essential to every vessel. In the more complex styles of rig square-sails predominate; chiefly because the fact of their being balanced upon the mast makes them comparatively easy to work.

All the wood-cuts in this Section are drawn to a scale of about $\frac{1}{100}$ of the real dimensions.

12. *Rig of Boats*.—Open boats are always fore-and-aft rigged, with one, two, or three moveable masts, and with or without a running bowsprit. They have seldom or never topsails, or other upper sails. Each mast carries a fore-and-aft sail of one or other of the kinds represented in Figs. 3, 4, 5, 6, and 7 of Article 6; these are called in their order, from the head, the *foresail*, *mainsail*, and *mizen* or *driver*; and when there is a bowsprit, a *jib* is carried on a stay between it and the foremast. When there is a driver, its clew is usually hauled out to a boom, or outrigger, projecting over the stern.

Fig. 10.

Fig. 11.



For example, Fig. 10 represents a gig about 20 feet long, with a single lug-sail; Fig. 11, a boat about 32 feet long, with a jib and three sprit-sails—fore, main, and mizen.

The following summary of proportions for the masts and sails of boats, is chiefly condensed from the examples given by Fincham:—

Base of sail = line of flotation ×	from 1·0 to 1·6
Leverage of sail = extreme breadth ×	" 1·2 to 1·4
Height of centre of effort above base of sail = } extreme breadth ×	" 0·8 to 1·2
Area of sail = line of flotation × extreme } breadth ×	" 1·4 to 2·5
Outboard length of bowsprit (if any) = line of } flotation ×	" 0·25 to 0·42

STATIONS FOR MASTS, in fractions of line of flotation from the middle:—

Boats with Two Masts—Fore and Main.

Mainmast,		Foremast,
from '03 abaft		from '34 afore
to '03 afore.		to '37 afore.

Boats with Two Masts—Fore and Mizzen.

Mizzenmast,		Foremast,
from '3 abaft		from '2 afore
to '5 abaft.		to '36 afore.

Boats with Three Masts.

Mizzenmast,	Mainmast,	Foremast,
about '5 abaft.	from '04 abaft	from '28 afore
	to '07 afore.	to '37 afore.

RAKE OF MASTS:—

Mizzenmast rakes aft.....	from 0 to 1 in 3
Foremast rakes aft.....	" 0 to 1 in 6
(Except in lateen-rigged boats, in which the foremast rakes forward from 1 in 12 to 1 in 3.)	

Mizzenmast usually rakes at the same rate with the stern-post.

PROPORTIONS OF MASTS:—

Length of mainmast (or of foremast where there is no mainmast) = extreme breadth of boat ×	from 2 to 3
Length of foremast = length of mainmast ×	" 0·8 to 1·0
Length of mizzenmast = length of mainmast ×	" 0·5 to 0·7

DISTRIBUTION OF THE AREA OF SAIL, per cent. of whole area:—

TWO SAILS.					
Mizzen,		Foresail,			Total.
from 26	+	74	=		100
to 30	+	70	=		100
THREE SAILS (WITH JIB).					
Mizzen,		Foresail,	Jib,		Total.
from 25	+	55	+	20	= 100
to 20	+	60	+	20	= 100
Mainsail,		Foresail,	Jib,		Total.
from 45	+	38	+	17	= 100
to 47	+	35	+	18	= 100
THREE SAILS (WITHOUT JIB).					
Mizzen,		Mainsail,	Foresail,		Total.
from 16	+	44	+	40	= 100
to 13	+	46	+	41	= 100
FOUR SAILS.					
Mizzen,		Mainsail,	Foresail,	Jib,	Total.
from 13	+	43	+	34	+ 10 = 100
to 16	+	38	+	32	+ 14 = 100

13. A *Lugger* is a small decked vessel equipped chiefly with the lug-sails already described in Article 6, and represented in

Fig. 12.



Fig. 12 represents a three-masted lugger, carrying a jib, three lug-sails (fore, main, and mizzen or driver), and three lug-topsails.

Fig. 3. The masts are usually three in number—mizzen, main, and fore—with a running or moveable bowsprit. Sometimes all three masts have separate topmasts carrying lug-topsails; sometimes the mainmast only is so fitted.

The following are proportions for this style of rig, taken from examples given by Fincham:—

Base of sail = line of flotation ×	about 1·6
Leverage of sail = extreme breadth ×	from 1·6 to 1·75
Height of centre of effort above base of sail = } extreme breadth ×	" 1·0 to 1·25
Area of sail = area of load-water section ×	" 3·8 to 4·0
= line of flotation × extreme breadth ×	" 2·5 to 3·0
Length of bowsprit out-board = line of flotation ×	about 0·4
Steeve (or upward slope) of bowsprit.....	about 1 in 24
Length of driver-boom or outrigger = line of } flotation ×	from '4 to '55

STATIONS FOR MASTS, in fractions of line of flotation from the middle:—

Mizzenmast,		Mainmast,		Foremast,
Abaft,		Abaft,		Afore,
from '4 to '44.		about '04.		about '4.

RAKE OF MASTS:—

Foremast rakes aft.....	from $\frac{1}{8}$ to $\frac{1}{4}$
Mainmast "	" $\frac{1}{8}$ to $\frac{1}{4}$
Mizzenmast "	" $\frac{1}{4}$ to $\frac{1}{2}$

PROPORTIONS OF MASTS:—

Length of mainmast, from heel to hounds = } extreme breadth of vessel ×	from 3 to 3·5
Length of foremast = length of mainmast ×	about 0·8
Length of mizzenmast = length of mainmast ×	" 0·6
Length of head of each lower mast = length of } topmast ×	" 0·25
Length of each topmast from heel to hounds = } lower mast ×	" 0·5
Length of each topmast pole-head = hounded } length of topmast ×	" 0·2

LENGTHS OF LOWER YARDS, in fractions of line of flotation (exclusive of yard-arms):—

Mizzen,		Main,		Fore,
about '35.		about '7.		about '6.
Length of each topsail-yard = length of lower } yard ×				from '65 to '85
Length of each lower yard-arm (or end of the } yard, which projects beyond the head of the } sail) = length of yard ×				about '04
Length of each topsail yard-arm = length of yard ×				" '1

Each of the yards is slung at about $\frac{1}{3}$ of its length from the forward or weather end, and *peaked* or sloped so that the lee-leech of each of the lower sails is from $\frac{1}{4}$ to $\frac{1}{2}$ deeper than the weather-leech. The topsail yards are parallel, or nearly so, to the lower yards.

A lugger is usually furnished with a second or smaller set of sails, of from '9 to '7 of the breadth of the first set, with yards to correspond.

The lugger-rig is considered advantageous for small vessels, because of its simplicity, and its spreading a great area of sail as compared with the moment of sail; but it is troublesome in tacking and other manœuvres, because of the necessity of shifting the sails to the lee side of the masts every time the vessel goes about, and is not well suited to large vessels.

The Chinese *junk* may be regarded as a modification of the lugger. The sails are stiffened transversely at intervals by bamboos, fixed to travellers or hoops that run upon the mast.

13 A. *Lateen-rigged vessels* (used chiefly in the Mediterranean) are equipped wholly or chiefly with lateen-sails, such as those shown in Fig. 4 of Article 6. They have one, two, or three masts, and are called by a variety of names; those most commonly used in English being *felucca* for one-masted, *galley* for two-masted, and *zebec* for three-masted vessels. The lateen rig is sometimes combined with the square rig, in various ways.

The following proportions are taken from examples given by Fincham:—

Base of sail = line of flotation ×	from 1·3 to 1·4
Leverage of sail = extreme breadth ×	" 1·2 to 1·5
Height of centre of effort above base of sail = } extreme breadth ×	" 0·8 to 1·0
Area of sail = area of load-water section ×	" 2·1 to 2·6
= line of flotation × extreme breadth ×	" 1·6 to 2·0

STATIONS FOR MASTS, in fractions of line of flotation from the middle:—

Mizenmast, Aft,	Mainmast, nearly amidships.	Foremast, Afore,
about 0·4.	...	about 0·4.

RAKE OF MASTS:—

Foremast rakes forward,	from $\frac{1}{8}$ to $\frac{1}{4}$
Mainmast,	upright.
Mizenmast rakes aft,	about $\frac{1}{4}$

PROPORTIONS OF MASTS:—

Length of mainmast, from heel to hounds = } extreme breadth of vessel ×	about 2·4
Length of foremast = length of mainmast ×	from '85 to '95
Length of mizenmast = length of mainmast × ...	about '6
Length of each mast-head = length of body of } mast ×	from '1 to '125

PROPORTIONS OF LATEEN YARDS:—

Length of main-yard = line of flotation ×	from '9 to 1, nearly.
Length of fore-yard = length of main-yard × ...	about '9
Length of mizen-yard = length of main-yard ×	" '45
Length of each weather yard-arm = length of } yard ×	from '01 to '014
Length of each lee yard-arm = length of yard ×	" '023 to '038

Each of the yards is slung at about $\frac{1}{10}$ of its length from its forward or weather end.

The lateen rig is as simple as the lugger rig; but it has not the same advantage as to spread of sail, and presents similar inconveniences in going about.

14. A *Cutter* (sometimes called, in the merchant service, a *sloop* or *smack*) is a one-masted vessel, whose general style of rig is shown in Fig. 13: *a* being the *jib*, set between the foremast and a running bowsprit; *b*, the *foresail*, which is a

Fig. 13.



triangular stay-sail; *c*, the main-sail, which is a gaff-sail, hanging from the gaff, and having its weather-leech bent to hoops on the mast: its clew is hauled out to the end of a boom. The mark \odot shows the centre of effort for the three sails before-mentioned. The *gaff-top-sail*, *d*, marked by dotted lines, is bent either to a short gaff, or to a yard like that of a lug-sail, which is hoisted on a pole topmast.

A cutter usually has a series of jibs, four or five in number, to suit different states of the wind. The largest is called the *main-jib*, the next the *second jib*, and so on. The smallest is of about half the dimensions of the largest each way. In order to suit the different sizes of jibs, the bowsprit has a large ring

or *traveller* upon it, to which the tack of the jib is hooked, and which can be hauled out or in to different distances. Sometimes also there is a second and smaller fore-sail, called the *storm fore-sail*. The main-sail is made to reef at the foot, the number of reefs being three or four; and for stormy weather, there is a sail of the same shape with the main-sail, and about one-third of the area, called the *try-sail*.

In addition to the before-mentioned sails, a cutter is often provided with a *balloon-jib*, or *jib top-sail* (*e*, Fig. 13), being a triangular sail hauled up to the topmast head, and with a *square-sail*, a *square-top-sail*, and sometimes a *square-topgallant-sail*, for running before the wind. The yard of the square-sail is called the *cross-jack-yard*; and the foot of that sail is often spread by means of a boom. Sometimes, instead of an ordinary square top-sail, there is a *half top-sail*, being a square-sail with a narrow head, and a foot which is half the breadth of the head of the square-sail, and is spread above the weather half of the cross-jack-yard. In light airs, the *ringtail* is spread beyond the leech of the main-sail.

The following proportions are founded upon Fincham's examples; the lower limits, as to area of sail, being taken from merchant sloops, and the higher from armed cutters:—

Base of sail = line of flotation ×	from 1·7 to 1·9
Leverage of sail = extreme breadth ×	about 1·5
Height of centre of effort above base of sail = } extreme breadth ×	about 0·9
Area of sail = area of load-water section ×	from 3·0 to 3·6
= line of flotation × extreme breadth ×	" 2·3 to 2·6

SAILS.

Noek of main-sail above water-line = extreme } breadth of vessel ×	from 2·1 to 2·6
Leech of main-sail = luff ×	" 1·4 to 1·6
Clew of main-sail abaft stern-post at water-line } = line of flotation ×	" '2 to '3
Foot of main-sail extends from clew to mast.	
Head of main-sail = foot of main-sail ×	" '6 to '75
Foot of fore-sail = distance from mast to stay ×	about '9
Luff of fore-sail = length of fore-stay ×	from '8 to '87
Leech of fore-sail = luff of fore-sail ×	about '8
Tack of jib afore stem at water-line = line of } flotation ×	from '5 to '6
= foot of second jib.	
Luff of jib = length of jib-stay ×	" '8 to '85
End of bowsprit extends beyond tack of jib,	8 or 9 inches.
Boom extends beyond clew of main-sail,	'08 × foot of sail.
Gaff extends beyond peak of main-sail,	'06 × head of sail.

BOWSPRIT, MAST, AND SPARS.

Length of bowsprit outboard = line of flotation ×	from 0·5 to 0·6
Steeve of bowsprit,	" 1 in 18 to 1 in 8
Mast afore middle = line of flotation ×	" '1 to '14
Rake of mast aft,	" $\frac{1}{4}$ to $\frac{1}{2}$
Length of mast from heel to hounds = extreme } breadth of vessel ×	" 2·6 to 2·8
Noek of main-sail below hounds = hounded } length ×	about '08
Length of lower mast-head = length of topmast ×	" '3
Length of topmast = length of lower mast × ...	" '67
Length of topmast pole-head = length of top- } mast from heel to hounds ×	from '17 to '30
Main boom = line of flotation ×	" '8 to '9
Main gaff = main boom ×	" '72 to '64
Gaff-top-sail-yard = main gaff ×	" '25 to '4
Half-top-sail-yard = gaff-top-sail-yard.	
Cross-jack yard = line of flotation ×	" '8 to '9
Square-top-sail-yard = cross-jack yard ×	about '7
Square-topgallant-sail-yard = cross-jack yard ×	" '5
Square-sail boom (to spread foot of square-sail) } = cross-jack yard ×	from '5 to '6
Ringtail boom = main boom ×	" '5 to '6

* "Cross-jack" is pronounced by seamen "Cro-jack."

The cutter-rig is the most favourable of all to the efficiency of the sails, especially in sailing near the wind, and in manœuvring; but the great size of the main-sail makes it impracticable to use it in large vessels; and in battle, or in a violent storm, its having only one mast exposes the vessel to too great a risk of being disabled.

15. A *Yawl*, or *Dandy*, has a mainmast and a running bowsprit with sails like those of a cutter; and in addition, there is a small mizenmast at the transom, rigged with a lug-sail or gaff-sail. The main-boom and foot of the main-sail are shortened, so as to clear the mizenmast; the other dimensions of the sails on the mainmast are the same as for a cutter, or nearly so. The following are ordinary proportions for the mizenmast and its sail:—

Length of mizenmast = mainmast ×	about .7
Driver-yard = main-gaff ×	" .7
Outrigger or fixed boom (for spreading clew of driver) } = outboard length of bowsprit,	nearly.
Area of larger driver = deficiency of main-sail as } compared with that of a cutter ×	2 nearly.
Area of smaller driver = area of larger ×	3 nearly.

The object of the yawl rig is to lighten the main-sail and its boom. It is much used for yachts.

16. A *Schooner* is a fore-and-aft-rigged vessel, with two masts, fore and main, or three masts, fore, main, and mizen. The bowsprit is either a running bowsprit in one piece, or a small standing bowsprit with a jib-boom. Each of the masts has a topmast, like that of a cutter. On the lower masts are spread gaff-sails, like a cutter's main-sail; but in general the aftermost of those alone (main-sail when there are two masts, driver when there are three) has a boom. The main-topsail, and the mizen-topsail (if any) are gaff-topsails; the fore-topsail and fore-topgallant-sail are usually square; and sometimes there is a square fore-sail, like that of a cutter. A jib top-sail may be set on the fore-topmast stay, which runs from the fore-topmast head to the jib-boom end; and a maintopmast-stay-sail on the maintopmast stay, which runs from the maintopmast head to the cap of the foremast. Sometimes the mainmast has a square-sail, top-sail, and topgallant-sail.

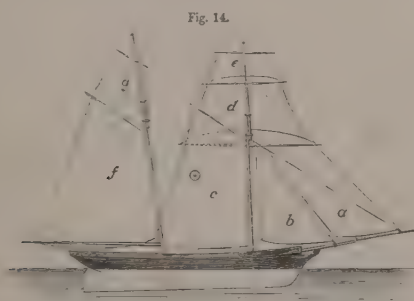


Fig. 14 shows the principal sails of a two-masted schooner: *a* being the jib; *b*, the fore-stay-sail; *c*, the fore-sail; *d*, the square fore-top-sail; *e*, the square fore-topgallant-sail; *f*, the main-sail; *g*, the main-gaff-topsail.

The following proportions, for two-masted schooners, are founded on Fincham's rules and examples:—

Base of sail = line of flotation ×	from 1.6 to 1.7
Leverage of sail = extreme breadth ×	about 1.75
Height of centre of effort above base } of sail = extreme breadth ×	" 1.12
Area of sail = area of load-water } section ×	" 3.6
= line of flotation × extreme breadth ×	" 2.7

Sails set—
Jib, Fore-sail,
Fore-sail, Main-sail,
Square foretop-sail,
Gaff-maintop-sail

STATIONS FOR MASTS, in fractions of line of flotation from the middle:—

Mainmast, Aft,	Foremast, Afore,
.05 to .11	.28 to .34
Rake of mainmast, aft,	from 1 in 6 to 1 in 4
Rake of foremast, aft,	from 1 in 10 to 1 in 4

SAILS.

Nock of main-sail above water = extreme } breadth ×	from .2 to 2.4
Nock of fore-sail above water = extreme breadth ×	" 1.8 to 2.3
Leech of main-sail and fore-sail = luff ×	" 1.2 to 1.3
(Inclination of main and fore gaffs, 25° to 30°.)	
Clew of main-sail abaft stern-post at water-line } = line of flotation ×	" .2 to .26
Foot of main-sail extends from clew to mainmast.	
Head of main-sail = foot of main-sail ×	" .5 to .6
Foot of fore-sail = foot of main-sail ×	" .7 to .85
Head of fore-sail = head of main-sail ×	" .75 to 1.0
Tack of jib afore stem on load-water line = line } of flotation ×	" .41 to .46
Foot of jib = line of flotation ×	" .32 to .35
Foot of square-foretop-sail = line of flotation ×	" .5 to .6
Head of foretop-sail = foot ×	" .6 to .9
Depth of foretop-sail = extreme breadth of vessel, nearly.	

BOWSPRIT, MASTS, AND SPARS.

Bowsprit from heel to cap = line of flotation ×	about .33
" length outboard = line of flotation ×	" .12
Jib-boom = line of flotation ×	" .4
Steeve of bowsprit,	from 1 in 6 to 1 in 4
Mainmast from heel to hounds = extreme } breadth ×	from 2.6 to 2.8
Foremast, from heel to hounds = mainmast ×	" .9 to .97
Nock of each gaff-sail below hounds of masts = } hounded length ×	about .045
Heads of lower masts = topmasts ×	from .3 to .4
Head of bowsprit = jib-boom ×	about 1/3
Topmasts, hounded = extreme breadth ×	from .83 to 1.0
Pole-heads of topmasts = hounded length ×	about .5
Main-boom = line of flotation ×	from .66 to .7
Main-gaff = main-boom ×	" .44 to .53
Fore-gaff = main-gaff ×	" .73 to 1.00
Fore yard = line of flotation ×	" .48 to .57
Fore-topsail-yard = fore-yard ×	" .7 to .75
Fore-topgallantsail-yard = fore-yard ×	" .42 to .48
Square-sail-yard (on mainmast), main-topsail- yard, and main-topgallant-yard, nearly equal to corresponding yards on topmast.	
Square-sail-boom = line of flotation ×	" .87 to .94

In a three-masted schooner, the bowsprit and fore and main masts, with their spars and sails, are of the same dimensions, or nearly so, with those of a two-masted schooner of the same size, except that the sails are sometimes taunter (that is, more deep and lofty); the height of the nock of the main-sail, above water, being as much as 2.6 times the extreme breadth; but the station of the mainmast is somewhat further forward, being .03 of the line of flotation abaft the middle, or thereabouts.

The mizenmast is about .36 or .37 of the line of flotation abaft the middle.

It rakes aft about 1 in 4 or 5; and the rake of the mainmast is made somewhat less.

The height of the nock of the driver, above the water-line, is about .65 of that of the nock of the main-sail.

Length of mizenmast from heel to hounds = mainmast ×	about .75
Head of mizenmast = topmast ×	from .3 to .4
Mizentopmast = maintopmast ×	about .9
Driver-boom = line of flotation ×	" .4
Driver-gaff = main-gaff ×	" .9

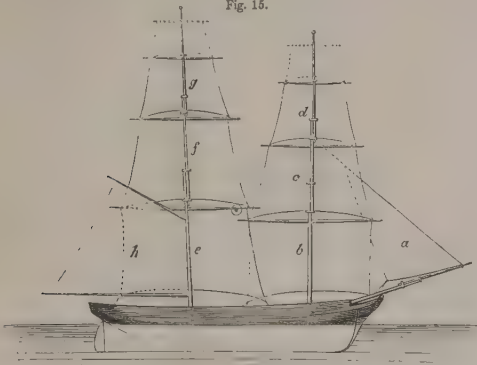
The three-masted schooner-rig spreads a proportionally greater area of sail than any other; being in some examples nearly five times the area of the load-water section.

17. A *Brig* is a two-masted, square-rigged vessel; each of the two masts having all the sails specified in Article 8 of this Division. There is a standing bowsprit, with a jib-boom and flying jib-boom.

Besides the square-sails, and the fore-and-aft sails on the bowsprit, a brig has also a large *gaff-mainsail*, called sometimes the *driver*, and a gaff-sail on the foremast, called the *fore-trysail*. (When the driver is bent to rings on a try-sail mast, the brig is called a *snow*.)

Fig. 15 shows the principal sails of a brig: *a* being the *jib*; *b*, the *fore-sail*, or *fore-course*; *c*, the *fore-top-sail*; *d*, the *fore-topgallant-sail*; *e*, the *main-course*; *f*, the *main-top-sail*; *g*, the *main-topgallant-sail*; *h*, the *driver*. These are the sails usually

Fig. 15.



taken into account in finding the centre of effort, which is marked thus, \odot ; the fore and main *royals*, which may or may not be included in that calculation, are shown by dotted lines above the topgallant-sails.

The following proportions are founded on Fincham's rules and examples:—

Sails set as above enumerated.	Base of sail = line of flotation \times	from 1.6 to 1.65
	Leverage of sail = extreme breadth \times	about 1.75
	Height of centre of effort above base of sail = extreme breadth \times	" 1.1
	Area of sail = area of load-water section \times	from 3.5 to 3.75
	= line of flotation \times extreme breadth \times	about 2.8

STATIONS FOR MASTS, in fractions of line of flotation from the middle:—

Mainmast, Aft,	Foremast, Afore,
from .14 to .15	from .81 to .83
The centres of the square-sails are about .05 of the extreme breadth afore the centres of the masts on which they are set.	
Rake of mainmast, aft,	from 1 in 16 to 1 in 13
Rake of foremast, aft,	" 0 to 1 in 50
Steeve of bowsprit,	" 1 in 3 to 1 in 2.8

SAILS.

Nock of driver is nearly on a level with, or a little below, the centre of effort of the jib, courses, driver, top-sails, and topgallant-sails.	
Clew of driver abaft stern-post at load-water-line = line of flotation \times	from .16 to .19
Foot of driver extends from clew to mainmast.	
Head of driver = foot \times	about .6
Leech of driver = luff \times	" 1.5
Tack of jib afore stem at load-water-line = line of flotation \times	" .45
Foot of jib = line of flotation \times	" .3
Luff of jib = length of jib-stay from fore-topmast head to jib-boom end \times	" .8
Leech of jib = luff \times	" .8

Height of head of main-top-sail above base of sail }
= height of centre of effort above base of sail \times } 2 nearly.

This height is divided into two nearly equal parts for the main-course and main-top-sail.

Depth of main-topgallant-sail = the same height \times from .30 to .34

Depth of fore-course = depth of main-course \times " .80 to .84

Head of main-course = line of flotation \times " .47 to .50

Head of main-top-sail = head of main-course \times about .7

Head of main-topgallant-sail = head of main-course \times " .5

Head of fore-course = head of main-course.

Heads and depths of fore-top-sail and fore-topgallant-sail = heads and depths of main-top-sail and main-topgallant-sail.

Depth of royals = depth of topgallant-sails \times

Heads of royals = heads of courses \times

The depth of canvas in a square-sail is less than the depth from centre to centre of the yards between which it is set, by about $\frac{1}{3}$ of that depth.

The extent to which sails are *roached*, or cut hollow at the foot, will be considered in a later chapter.

BOWSPRIT, MASTS, AND SPARS.

Height of hounds of lower masts above heads of courses }
= depth of top-sails \times

Projection of jib-boom end beyond tack of jib = foot of jib \times

Length of jib-boom, 23 inches less than foot of jib.

Projection of outer end of bowsprit beyond heel of jib-boom = length of jib-boom \times

Heel of bowsprit steps on bowsprit partners, before the foremast.

Flying jib-boom = jib-boom \times

Projection of peak of gaff beyond peak of driver = head of driver \times at least

Additional length of peak for displaying signals, from 3 to 6 feet.

Hounded length of topmast = depth of top-sail, nearly.

Hounded length of topgallantmast = depth of topgallant-sail, nearly.

Royal pole = topgallantmast hounded \times

Sky-sail pole or signal pole = royal pole \times

Length of each lower-mast head = hounded length of topmast \times

Length of each topmast head = hounded length of topgallantmast \times

Each lower, topgallant, and royal yard arm = head of sail \times

Each topsail-yard arm = excess of half breadth of sail at the middle of its depth above half breadth at the head + $\frac{1}{10}$ of head of sail.

Spritsail-yard (across the bowsprit) = main-top-sail-yard.

Main-trysail-gaff = main-gaff \times

Fore-trysail-gaff = main-gaff \times

Swing-booms (for feet of lower studding-sails) = lower yards \times

The upper sails and spars of the two masts of a brig are made of equal dimensions, in order that the same spare spars and sails may answer to replace those of either mast.

The rig of a brig is useful chiefly for merchant vessels, because of its dividing the whole area of sail into small parts, which can be handled by a small crew.

18. A *Brigantine*, or *Hermaphrodite*, has the bowsprit and foremast of a brig, and the mainmast of a schooner, each with its proper station, proportions, rig, and sails. Vessels of this kind are used in the merchant service, with a view to combining to a certain extent the advantages of the square rig and fore-and-aft rig.

18A. A *Ketch* has two masts—a large square-rigged mainmast, like that of a brig, and a small mizenmast, like that of a three-masted schooner. The bowsprit and jib-boom are like those of a brig.

The stations for the masts are nearly as follows, in fractions of the load-water-line from the middle:—

Mizenmast, Abaft, about $\frac{1}{4}$	Mainmast, Afore, about $\frac{1}{11}$.
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This style of rig is now little used. According to Fincham, it spreads a smaller area of sail, in proportion to the moment, than any other.

19. A *Ship* is a vessel with three square-rigged masts, each provided with the series of sails mentioned in Article 8, a bowsprit and jib-boom, with jibs and stay-sails, and a gaff-sail on the mizenmast, called the *spanker* or *driver*. To show the names given to the masts, spars, and sails of a ship, reference may be made to Plate $\frac{2}{4}$ (the rigging-plan of the *Formby*):—

MASTS AND SPARS.

A, bowsprit.	with a special method of reefing top-sails.)
B, jib-boom.	Q, fore-topsail-yard, or upper fore-top-sail-yard.
C, flying jib-boom.	R, fore-topgallant-yard.
D, dolphin-striker.	S, fore-royal-yard.
E, foremast.	T, main-yard.
F, fore-topmast.	U, lower main-topsail-yard (see P, above).
G ₁ , fore-topgallantmast. } One spar.	V, main-topsail-yard, or upper main-topsail-yard.
G ₂ , fore-royal pole. }	W, main-topgallant-yard.
H, mainmast. }	X, main-royal-yard.
I, main-topmast. }	Y, Cross-jack (or Cro'-jack) yard.
J ₁ , main-topgallantmast. } One spar.	Z, Lower mizen-topsail-yard (see P, above).
J ₂ , main-royal pole. }	AA, mizen-topsail-yard, or upper mizen-topsail-yard.
K, main-skysail pole, or signal pole. (Some ships have a skysail pole or signal pole on each mast.)	BB, mizen-topgallant-yard.
L, mizenmast.	CC, mizen-royal-yard.
M, mizen-topmast.	DD, main-trysail-gaff. (Some ships have a trysail-gaff on the foremast also.)
N ₁ , mizen-topgallantmast. } One spar.	EE, spanker-boom, or driver-boom.
N ₂ , mizen-royal pole. }	FF, spanker-gaff, or driver-gaff.
O, fore-yard.	
P, lower fore-topsail-yard. (Lower topsail-yards are of recent introduction, and are used in some ships only, in connection	

The following spars are not seen in the Plate.

In some ships, there is a *spritsail-yard*, crossing below the bowsprit a short way abaft the dolphin-striker; but in the present example that yard is not used. It does not carry a sail, but is used only for securing the rigging of the jib-boom and flying jib-boom. Sometimes instead of one spritsail-yard crossing the bowsprit horizontally, there are a pair of spars pointing obliquely downwards at opposite sides of it: these are called *spritsail-gaffs*.

Near the cat-heads, at the bow of the ship, two spars project, called the *boomkins* or *bumpkins*, for hauling out the weather-tack of the fore-sail in sailing near the wind. Sometimes, in small ships, the foot of the fore-sail is spread by the *fore-boom*, or *bentick-boom*.

The spanker and try-sails are sometimes bent to rings or hoops running on *trysail-masts*, parallel to, and close abaft the lower masts, to which they are fixed at the head and heel.

The *mizen-trysail-gaff*, smaller than the spanker-gaff, is used in stormy weather.

The *studding-sail booms*, when rigged out, project from the arms of the fore and main lower yards, and fore and main top-sail-yards, to which each of them is secured by passing through a pair of rings called *boom-irons* (to be described in detail in a later chapter). Each of those booms serves to spread the foot of a studding-sail, and is named from that studding-sail; thus,

the topgallant-studdingsail booms are carried by the topsail yard-arms, and the topmast-studdingsail booms by the lower yard-arms. The feet of the lower studding-sails are spread by means of booms called the *lower studding-sail booms*, or *swing-booms*. The heads of the studding-sails are bent to *studding-sail yards*, which are slung from the studding-sail booms, and from the fore and main topgallant yard-arms. For the reason stated in Article 7, there are no studding-sails on the mizenmast.

The *ringtail boom* is rigged out like a studding-sail boom, at the end of the spanker boom; and the *ringtail yard* is slung from the peak.

The *ensign-staff* is a pole fixed to the taffrail, and raking aft over the stern, for displaying the *ensign*, *jack*, or national flag of the ship, when the spanker is not bent. At other times the ensign is run up to the peak of the spanker-gaff, as shown in the Plate.

SAILS.

a, fore-staysail.	g, lower } main-topsail.
b, fore-topmast-staysail.	r, upper } (See h and i, above.)
c, inner jib.	s, main-topgallant-sail.
d, outer jib. (Most ships have only one jib, of an area about equivalent to the combined effective area of the two jibs in the present example.)	t, main-royal.
e, fore-topgallant-staysail.	u, main-trysail, or main-spencer. Sometimes there is a trysail or spencer on the foremast also. Sometimes the place of the main-trysail is occupied by the mizen-staysail.
f, flying jib.	v, mizen-topmast-staysail.
g, fore-sail, or fore-course.	w, mizen-topgallant-staysail.
h, lower } fore-topsail; being equivalent to an ordinary fore-topsail divided into two parts of equal depth.	x, mizen-course, or cro'-jack (seldom used).
i, upper }	y, lower } mizen-topsail.
j, fore-topgallant-sail.	z, upper } (See h and i, above.)
k, fore-royal.	aa, mizen-topgallant-sail.
l, main-staysail.	bb, mizen-royal.
m, main-topmast-staysail.	cc, spanker or driver (instead of which the mizen-trysail is hoisted in stormy weather).
n, main-topgallant-staysail.	
o, main-royal-staysail.	
p, main-sail, or main-course.	

The following additional sails are not seen in the Plate:—

Sky-sails, *sky-scrapers*, or *flying kites*, being small square-sails above the royals.

Studding-sails on the foremast and mainmast, already mentioned. (Lower studding-sails on the mainmast are seldom used.)

The *ringtail*, already mentioned, set beyond the leech of the spanker.

Water-sails, set in very light airs and smooth water, below the lower studding-sail booms.

The standing and running rigging, shown on the same Plate, will be explained in a later chapter.

In finding the centre of effort and moment of sail of a ship, it is usual to take the following sails only into account—*jib*, *fore and main courses*, *driver*, *three top-sails*, and *three top-gallant-sails*.

The following dimensions and proportions are founded chiefly on the examples and rules of Fincham, and on some other examples of more recent date:—

Base of sail = line of flotation ×	{ from 1·6 in short full ships to 1·35 in long fine ships.
Leverage of sail = extreme breadth ×	{ from 1·75 to 2·00
Height of centre of effort above base of sail = extreme breadth ×	{ " 1·0 to 1·3
Area of sail = area of load-water section ×	{ " 3·0 to 3·9
= line of flotation × extreme breadth ×	{ " 2·2 to 2·9

Sails set as above enumerated.

STATIONS FOR MASTS, in fractions of line of flotation from the middle:—

Mizenmast, Abaft,	Mainmast, Abaft,	Foremast, Afore,
from 0·4 to 0·3 ...	from 0·08 to 0·03 ...	from 0·4 to 0·3

The centres of the square-sails on a mast are about '05 of the extreme breadth before the station of the mast.

Rake of foremast, aft,.....	from 0 to 1 in 36
Rake of mainmast, aft,.....	" 0 to 1 in 12
Rake of mizenmast, aft,.....	" 0 to 1 in 12
Steeve of bowsprit,.....	" 1 in 3 to 1 in 2

SAILS.

Noek of driver above base of sail = height of centre of effort above base of sail X.....	from '6 to '75
Clew of driver abaft stern-post at load-water-line = line of flotation X.....	" '23 to '05
Foot of driver extends from clew to mizenmast, or mizen-trysail-mast.	
Head of driver = foot X.....	from '7 to '75
Leech of driver = luff X.....	{ from 1·5 (for a deep luff) to 2·0 (for a short luff).
Tack of jib afore stem at load-water-line = line of flotation X.....	from '4 to '3
Foot of jib = line of flotation X.....	from '27 to '3
Luff of jib = length of jib-stay from foretopmast head to jib-boom end X.....	about '8
Leech of jib = luff X.....	" '8
Height of head of main-topsail above base of sail = height of centre of effort above base of sail X.....	from 2·0 to 1·75
Depth of main-topsail = height of centre of effort above base of sail X.....	" 1·0 to 0·9
Depth of main-course determined from the position of the foot of the main-topsail.	
Depth of fore-course = depth of main-course X.....	" 0·9 to 0·75
Depth of mizen-course determined by cro' jack yard being close above nock of driver.	
Depth of fore-topsail = depth of main-topsail X.....	" 0·87 to 1·0
Depth of mizen-topsail = depth of main-topsail X.....	" 0·75 to '65
Depths of topgallant-sails = depths of top-sails X.....	" 0·5 to 0·6
Depths of royals = depths of topgallant-sails X.....	about '67
Head of main-course = line of flotation X (in short ships).....	{ from '48 to '52
(In long ships).....	" '36 to '48
Head of fore-course = head of main-course X.....	" '85 to 1·0
Head of mizen-course = head of main-course X.....	" '65 to '75
Heads of top-sails = heads of courses X.....	" '7 to '8
Heads of topgallant-sails = heads of courses X.....	" '4 to '6
Heads of royals = heads of courses X.....	" '3 to '45
Breadth of a studding-sail = breadth of sail beside which it is set X.....	" '5 to '6

Depth of canvas in a square-sail about $\frac{3}{16}$ part less than the total depth from centre to centre of the yards between which it is set.

The *roaching* or hollowing of sails at the foot, and other details, will be treated of in a later chapter.

DISTRIBUTION OF THE BASE OF SAIL, per cent. :—

Driver,	Main-course,	Fore-course,	Jib,	Total.
from 22	+ 31	+ 28	+ 19	= 100
to 20	+ 29	+ 29	+ 22	= 100

BOWSPRIT, MASTS, AND SPARS.

Height of hounds of lower masts above heads of courses and nock of driver = depth of main-topsail X.....	about $\frac{1}{8}$
Projection of jib-boom end beyond tack of jib = foot of jib X.....	" '03
Length of jib-boom about 28 inches less than foot of jib.	
Projection of outer end of bowsprit beyond heel of jib-boom = length of jib-boom X.....	$\frac{3}{8}$
Heel of bowsprit steps on bowsprit partners, afore the fore-mast.	
Flying jib-boom = jib-boom X.....	from 0·6 to 1·2
Projection of peak of gaff beyond peak of driver = head of driver X.....	at least about '045
Additional length of peak for displaying signals.	from 3 to 6 feet
Hounded length of topmast = depth of topmast nearly.	
Hounded length of topgallantmast = depth of topgallant-sail nearly.	
Royal pole = topgallantmast hounded X.....	about '7
Skysail pole or signal pole = royal pole X.....	" $\frac{1}{3}$

Length of each lower-mast head = hounded } about '3	
length of topmast X.....	
Length of each topmast-head = hounded length of topgallantmast X.....	" '24
Each lower, topgallant, and royal yard-arm = head of sail X.....	" '05
Each topsail yard-arm = excess of half-breadth of sail at the middle of its depth above half-breadth at the head + $\frac{1}{10}$ of head of the topsail.	
Spritsail-yard = fore-topsail-yard; or	
Each spritsail-gaff = fore-topsail yard X.....	" $\frac{1}{2}$
Mizen-trysail-gaff = driver-gaff X.....	" '3
Main-trysail-gaff = driver-gaff X.....	" '6
Swing-booms, each = main-yard X.....	" '6
Fore-boom = fore-yard	

It may be observed that in one of the limiting cases given in the above statement, the upper sails and spars of the foremast and mainmast are of equal dimensions, the only difference being in the depths of the courses. The object of this in ships (as in brigs), is to enable the same spare spars and sails to answer for either of those masts. It is more practised now than it was in former times.

20. A *Barque* has three masts; the foremast and mainmast being square-rigged, like those of a ship, and the mizenmast fore-and-aft-rigged, like that of a schooner. The stations, dimensions, sails, and rig for the mainmast, foremast, and bowsprit, are the same as for a ship, except that the foremast, as well as the mainmast, has almost always a try-sail, and those try-sails are somewhat larger for a barque than for a ship.

The foot of the driver, and the driver-boom, may be laid-off as for a ship; but that sail is deeper than a ship's driver.

The following are the proportions peculiar to a barque :—

Noek of driver, nearly on a level with the centre of effort.	
Head of driver = foot X.....	from '63 to '75
Depth of gaff-topsail = depth of main-topsail X.....	" '9 to 1·0
Mizentopmast = maintopmast X.....	" '85 to '9
Main-trysail-gaff = driver-gaff X.....	" '63 to '71
Fore-trysail-gaff = driver-gaff X.....	" '66 to '78
Mizen-topsail-gaff = driver-gaff X.....	" '25 to '5

20A. *Vessels with more than three masts* are uncommon, and are met with chiefly amongst steam-vessels, because of their great proportionate length, which sometimes makes it difficult to spread the required area of sail on three masts only. In a four-masted ship, the second or bonaventure mizenmast is always fore-and-aft rigged, like that of a barque.

21. *Steam-vessels*.—The propelling action of canvas and steam combined, has been considered in the First Division, Article 184.

The fore-and-aft rig is preferable to the square-rig for steamers, on account of the comparatively small resistance that the former style of rig meets with in steaming against the wind with sails furled. Hence the square-rig is in general used in steamers only partially, the commonest rig being that of two-masted or three-masted schooners; and even the largest steamers, and those most fully covered with canvas, are almost always rigged as schooners, brigantines, or barques, with two, three, or four masts, and seldom as full-rigged ships or as brigs.

A steam-vessel that is intended to make long voyages under canvas alone, or with little assistance from steam, is fitted with masts and sails equal to those of a sailing vessel of the same dimensions and rig; and such is usually the case with sea-going war steamers, how powerful soever their engines may be, with many steam-yachts, and with merchant vessels having "auxiliary" screw engines of small power. The chimneys are usually between the foremast and mainmast, being the place where they

interfere least with the working of the sails. See, for example, H.M.S. *Warrior*, Plate $\frac{B}{7}$.

Steam-vessels in which steam is to act always as the principal propelling power, have in general a less area of sail than sailing vessels of the same dimensions. The *base of sail*, in particular, is proportionally short, as compared with the length of the line of flotation. Its ratio to that line varies very much; but the ordinary values may be estimated as ranging from 1 to 0.5. This shortening of the base of sail is in general produced chiefly by there being a gap in the middle of it, over the boiler-room; and sometimes also by the bowsprit being very short, or altogether wanting, as is often the case in a vessel with a long fine entrance. In many steam-vessels, the base of sail is nearly equal to that of a sailing vessel of a length equal to from $3\frac{1}{2}$ to 5 times the extreme breadth of the steamer.

The shortened base of sail is distributed amongst the feet of the several sails nearly as in sailing vessels.

The *mean height of sail*, in many steamers, is as great proportionally to the extreme breadth of the vessel as it is in sailing vessels; and such is very generally the case in screw-steamers, where an important use of the sails is to steady the vessel in a seaway; and their power of doing so depends on their moment. In other examples, found chiefly amongst the swifter class of paddle-steamers, the mean height of sail is considerably less than in a sailing vessel of the same extreme breadth; being sometimes little more than half.

Vessels which always use steam as the chief means of propulsion are seldom provided with those sails which are serviceable in light winds only, such as flying kites, studding-sails, and royals. Some, indeed, are without upper sails, and have *stump masts*; that is, lower masts without tops.

The rigging plan of the *Hope*, Plate $\frac{G}{9}$, may be taken as an extreme example of a swift paddle-steamer having sails much smaller in both dimensions than those of a sailing vessel of the same length and breadth. She is rigged as a two-masted

schooner, without a bowsprit; and the following are the spars and sails:—

A, foremast.	a, fore-staysail.
B, foretopmast.	b, jib.
C, mainmast.	c, jib-topsail.
D, maintopmast.	d, fore-sail.
E, fore-gaff.	e, fore-gaff-topsail.
F, fore-gaff-topsail-yard.	f, main-staysail.
G, main-boom.	g, main-sail.
H, main-gaff.	h, main-gaff-topsail.
I, main-gaff-topsail-yard.	

The principal proportions are as follows:—

Line of flotation = extreme breadth \times	8.0
Base of sail = line of flotation \times	0.6
Depth of centre of lateral resistance below water = extreme breadth \times	0.114
Height of base of sail above centre of lateral resistance = extreme breadth \times	0.57
Leverage of sail = extreme breadth \times	1.23
Mean height of sail (centre of effort above base) = extreme breadth \times	0.66
Area of sail = line of flotation \times extreme breadth \times	0.57
Centre of effort is nearly at middle of line of flotation.	
Distance between centres of head-sail and after-sail = line of flotation \times62
Centre of head-sail afore middle = line of flotation \times28
Centre of after-sail abaft middle = line of flotation \times34
Proportionate areas of—	
After-sail : Head-sail	
: : 14 : 17 nearly.	
Distance of each mast from middle = line of flotation \times	0.28

The first instance of a ship with six masts is presented by the *Great Eastern*. She has no bowsprit; and the rig and order of the masts are as follows:—

Foremast—fore-and-aft rigged.
Second mast—square rigged.
Third mast—square-rigged.
Fourth mast—fore-and-aft rigged, but capable of being fitted with square sails also.
Fifth mast—fore-and-aft rigged.
Sixth mast—fore-and-aft rigged.

For details as to the *Great Eastern*, reference must be made to Mr. Scott Russell's treatise on "Naval Architecture."

CHAPTER II.

OF MASTS AND SPARS.

SECTION I.—MATERIALS FOR MASTS AND SPARS.

22. The *timber* best suited for masts and spars consists of the stronger kinds belonging to the class of *Pine-wood*, or that produced by coniferous trees, such as pine, fir, larch, cowrie, &c.; because such timber combines strength with flexibility and lightness, and is to be had in long straight pieces. It has already been described in the Fourth Division, Articles 14, 15, and 16. The heavier and more resinous kinds, as red and yellow pine, are the best for lower masts and topmasts, and the larger yards; the lighter and less resinous kinds, as spruce or deal, are the best for the loftier and lighter spars, as topgallant masts and yards, studding-sail yards and booms, &c.

Pieces of timber suited for being made into masts and spars

are called *sticks*. Special care is used in their inspection, to see that they are sound. They are divided into squared sticks, called *inch-masts*, which are described according to the number of inches in the side; and round sticks, described according to their girth at the butt, in *hands* of four inches, and called *hand-masts* if the girth is not less than six hands, and *spars* if the girth is less than six hands. Spars are further subdivided as follows:—

Name.	Girth at the butt.
Cant spars,	from 6 to 5 hands.
Barling spars,	" 5 to 4 "
Boom spars,	" 4 to 3 "
Middling spars,	" 3 to 2 "
Small spars,	" 2 to 1 "

Immersion in mud is considered to be the best way of preserving mast-timber until it is wanted for use.

23. *Iron and Steel* are used for making tubular masts, bowsprits, and yards, in the form of plates, angle-bars, and rivets; as to the strength and quality of which, see Division III., Article 75, and Division IV., Article 7; and also the Rules of Lloyd's and of the Liverpool Registry, quoted in the Appendix to Division III. According to the rules of the Liverpool Registry, steel for making yards is treated as being stronger than iron in the ratio of 4 to 3.

SECTION II.—FIGURES AND DIMENSIONS OF MASTS AND SPARS.

24. *Principal Diameters of Masts, Bowsprits, and Jib-booms.*—

The bending load to which a mast is exposed is proportional to the area of canvas set upon it, which is roughly proportional to the square of the length of the mast; and the leverage with which that load acts is roughly proportional to the length of the mast; so that the bending moment may be roughly estimated as varying nearly as the cube of the length of the mast. The moment of resistance of the mast is proportional to the cube of its diameter (see Division III., Article 46, Rule IV.), and therefore the diameters of similarly situated masts ought to bear a nearly constant proportion to their lengths; and such is the rule followed in practice: the proper proportions in different cases having been ascertained by long experience.

The greatest diameter of a mast, bowsprit, yard, or any spar, is called the *given diameter*, and bears proportions to the length which are exemplified in the following table. The *end diameters* bear certain proportions to the given diameter.

	Position of given Diameter.	Ratio of given Diameter to Length from Heel to Hounds.	Ratios of end Diameters to given Diameters.
LOWER OR STANDING MASTS:—			
Ships and brigs,.....	{ At the partners of the wedge-ing deck, }	... $\frac{1}{4}$ to $\frac{1}{5}$...	{ Head, .67 to .75 Hounds, .75 to .80 Heel, .83
Schooners,.....	do.	... $\frac{1}{5}$ to $\frac{1}{4}$...	{ Head, .5 to .67 Hounds, .6 to .75 Heel, .83
Cutters,.....	do.	... do. ...	{ Head, .67 to .83 Hounds, .8 to .86 Heel, .83
Luggers,.....	do.	... do. ...	{ Head, .58 Hounds, .75 Heel, .83
Lateen rig,.....	do.	... $\frac{1}{5}$ to $\frac{1}{4}$...	{ Head, .75 Hounds, .83 Heel, .83
TOPMAST:—			
Square rigged,	{ At the cap of the lower mast, }	... $\frac{1}{5}$ to $\frac{1}{4}$...	{ Head, .7 Hounds, .8
Fore-and-aft rigged,...	do.	... $\frac{1}{5}$ to $\frac{1}{4}$...	{ Head, .5 Hounds, .7
TOPGALLANT-MAST,.....	{ At the cap of the topmast, }	... $\frac{1}{5}$ to $\frac{1}{4}$...	{ Pole, .5 to .55 Hounds, .8
BOWSPRIT:—			
Ships and brigs,.....	At the bed,	... $\frac{1}{5}$ to $\frac{1}{4}$...	{ Outer end, .67 Heel, .83
Cutters and schooners,	do.	... do. ...	{ Outer end, .67 Heel, 1.0
JIB-BOOM,.....	{ At the cap of the bowsprit, }	... $\frac{1}{5}$...	{ Outer end, .67 to .75 Inner end, 1.0
FLYING JIB-BOOM,	{ At the outer end of the jib-boom, }	... $\frac{1}{5}$...	{ Outer end, .67 Inner end, .75

25. *Principal Diameters of Yards, Booms, and Gaffs.*—The part of a yard at or near the middle, by which it is slung, is called the *slings*; the two endmost parts, projecting beyond the head of the sail, the *arms*; and the parts intermediate between the slings and the arms, the *quarters*. The lower or foremost

end of a gaff is called the *throat*; the upper or aftermost part, which projects beyond the sail, the *peak*; and the part intermediate between the throat and the peak, the *quarters*.

	Position of given Diameter.	Ratio of given Diameter to Length.	Ratios of end Diameters to given Diameters.
SQUARE YARDS:—			
Lower,.....	At slings, ...	$\frac{1}{4}$ to $\frac{1}{5}$...	Arms, .5
Topsail,.....	do. ...	$\frac{1}{5}$ to $\frac{1}{4}$...	" .5
Topgallant, royal, and studding sail,)	do. ...	$\frac{1}{5}$ to $\frac{1}{4}$...	" .5
LUG-SAIL YARDS,.....	do. ...	$\frac{1}{5}$ to $\frac{1}{4}$...	" .5
LATEEN YARDS,.....	do. ...	$\frac{1}{5}$ to $\frac{1}{4}$...	{ Fore end, .33 After end, .67
GAFFS:—			
For drivers, and fore-and-aft fore and main sails,)	Near throat, ...	$\frac{1}{5}$ to $\frac{1}{4}$...	Peak, 0.5 to 0.6
For trysails,.....	do. ...	$\frac{1}{5}$ to $\frac{1}{4}$...	" 0.6
BOOMS:—			
Main-booms of cutters, schooners, and brigs, and driver-booms,.....	{ At sheet or taffrail, or about one-third of length from after end of boom, }	$\frac{1}{5}$ to $\frac{1}{4}$...	{ Fore end, .67 After end, .75
Square-sail booms,.....	At middle, ...	About $\frac{1}{5}$...	From .75 to .67
Studding-sail booms and ring-tail booms,)	{ Throughout middle third of length, }	$\frac{1}{5}$ to $\frac{1}{4}$...	" .75 to .67

If the working modulus of stress on timber be taken at 1000 lbs. on the square inch, and the average area of sail spread on a yard as equal to the square of the length of the yard, it is easily deduced from the principles of Division Third, Article 46, that the diameters of yards used in practice are adapted to the following bending loads, in lbs. per square foot of canvas:—

Diameter = $\frac{1}{4}$ length,	432 lbs. per square foot.
" $\frac{1}{5}$ "	250 " "
" $\frac{1}{6}$ "	128 " "

The bending load upon a yard may be many times greater than the direct pressure of the wind on the sail, owing to the position and form of the sail.

From the principles of Article 48 of the Third Division, it appears that the longitudinal sections of the theoretical figures of uniform strength for yards and booms, and the dimensions of the frusta of cones which approach nearest to those figures, are as follows:—

Index of the power of the distance from the end, to which the diameter of the theoretical figure is proportional,.....	Yards.	Booms.
.....	$\frac{2}{3}$	$\frac{1}{3}$
Proportion of the smallest to the greatest diameter, in the conic frustum which approaches nearest to the theoretical figure,)	$\frac{1}{3}$	$\frac{2}{3}$

Hence it appears, that the smaller diameters used in practice are always greater than those which would be necessary if the strength of the material were uniform throughout the length of the yard or boom. The reason is, that the tapered part of the yard or boom is more or less grain-cut, and consequently weakened.

26. *Tapering of Masts and Spars.*—The diameters of masts and spars at the *quarters*, or parts intermediate between the greatest and smallest diameters, are not regulated by any theoretical principle, but are merely laid off so as to give a fair convex curvature to the outline. The distance from the given diameter to the small diameter is divided into four equal intervals, and the points of division are called respectively the *first*, *second*, and *third quarters*. In a lower mast, the points of division between the partners and the heel are called the *lower quarters*, and those between the partners and the hounds the

upper quarters. Then dividing the whole taper, or difference between the given diameter and the small diameter, into sixteen equal parts, those parts are to be thus distributed, in order to give the fairest possible curvature:—

Diameter at first quarter = given diameter — $\frac{1}{16}$ whole taper.
 " second " = " — $\frac{1}{8}$ "
 " third " = " — $\frac{3}{16}$ "

Mast-makers often use the fractions $\frac{1}{16}$, $\frac{1}{8}$, and $\frac{1}{4}$, instead of $\frac{1}{16}$, $\frac{1}{8}$, and $\frac{1}{4}$; but the curvature which they thus obtain is less fair.

From the third quarter to the hounds, a mast is tapered the fore-and-aft way only, its thwartship diameter throughout that division being kept uniform. The object of this is to provide the projections called *hounds*, below the lower end of the head, for supporting the trestle-trees and other framing.

27. *Thickness of Iron and Steel Masts and Spars*.—The following rule is a consequence of the principles stated in the Third Division, Article 46, Rule IV., Examples III. and VIII. of the Table:—

To find the thickness of the shell of a hollow iron or steel mast or spar, which shall be equally strong with a solid wooden mast or spar of the same diameter: Divide the diameter by eight times the ratio in which the working modulus of strength for the iron or steel is greater than that for the wood.

In estimating the working modulus of strength for hollow iron or steel masts or spars, regard must be had to the fact, that thin tubes of iron or steel usually give way to a bending load by buckling at the compressed side.

If the working modulus for iron under these circumstances be estimated at $7\frac{1}{2}$ times, and that for steel at 10 times, the working modulus for wood, the following results are obtained:—

$$\begin{aligned} \text{Iron,.....} & \frac{\text{thickness}}{\text{diameter}} = \frac{1}{8 \times 7\frac{1}{2}} = \frac{1}{60}; \\ \text{Steel,.....} & \frac{\text{thickness}}{\text{diameter}} = \frac{1}{8 \times 10} = \frac{1}{80}. \end{aligned}$$

The following are the proportions prescribed by Rule 21 and Table No. 8 of the Liverpool Registry (for which see the Appendix to the Third Division).

	Thickness, Diameter.
Iron masts,.....from	$\frac{1}{8}$ to $\frac{1}{4}$
Iron yards,.....	$\frac{1}{8}$ to $\frac{1}{4}$
Steel yards,.....	$\frac{1}{8}$ to $\frac{1}{4}$

SECTION III.—CONSTRUCTION AND FITTING OF MASTS AND SPARS.

28. *Heels and Steps—Bowsprit Partners*.—The heel of a wooden lower mast is formed into a rectangular tenon, measuring about two-thirds of the given diameter athwartships, and one-half of the given diameter fore-and-aft. This fits into a mortise of the same shape and dimensions in the top of the *step*—a piece of timber of siding and moulding nearly equal to the given diameter of the mast, and of a length equal to about twice the diameter of the mast, if the step rests on the keelson; or equal to once or twice the room and space of the deck-beams, if it rests on a deck. Fore and main steps always rest on and are coaked and bolted to the keelson; the mizen step sometimes rests on the keelson, and sometimes on two or three of the lower deck-beams; and in the latter case, those beams should be well supported from below by stanchions or pillars.

The heel of a wooden standing bowsprit is formed into a rectangular tenon, measuring about 0.6 of the given diameter athwartships, and 0.67 up and down; it abuts against and is

mortised into a piece of timber called the *bowsprit partners*, of siding equal to the given diameter of the bowsprit, and moulding equal to from one-half to two-thirds of that diameter. The bowsprit partners stands raking aft, so as to be at right angles to the bowsprit; its lower end is fastened to and supported by a beam of the deck next below the heel of the bowsprit, and its upper end bears against and is fastened to a beam of the deck next above, which is usually the upper deck. (For an example of the position of the bowsprit partners, see Plate $\frac{7}{7}$.)

The step of an iron or steel mast usually consists of a plate of iron or steel of from $1\frac{1}{2}$ times to double the diameter of the heel of the mast, resting on and bolted or rivetted to the keelson and the neighbouring floors. Upon its upper side is a circular rising ledge of L-shaped or T-shaped bar, rivetted to the plate, and inclosing a circular cavity into which the heel of the mast is fitted, and is kept steady by wooden wedges driven round it. (See Plates $\frac{8}{7}$ and $\frac{9}{7}$.) As in the case of wooden masts, the mizen step sometimes rests on the lower deck-beams. (See Plate $\frac{9}{7}$.) Iron masts are prevented from rotating, sometimes by having a square tenon on the heel, and sometimes by being bolted to the ledge of the step.

As to mast partners, see the Fourth Division, Article 40. It may be added, that iron or steel mast partners usually consist of a plate spanning over two or three deck-beams, and strengthened on the under side by fore-and-aft ribs or carlings of the same depth and thickness with the beams, or nearly so: in the plate is a circular hole of a size a little larger than the mast, and surrounded by a rising ledge or flange. (See Plates $\frac{8}{7}$ and $\frac{9}{7}$.)

The masts are usually wedged in the uppermost complete deck, which is called the *wedging-deck*; and it is here that the mast has the greatest or given diameter.

29. *Timber masts* are distinguished into *single-tree masts*, and *built or made masts*.

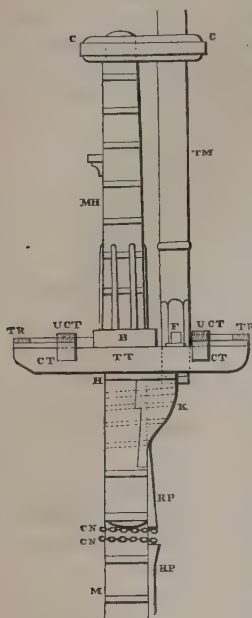
The stick for a single-tree mast should be about $\frac{1}{8}$ part longer than the total given length of the mast from heel to cap, and from 1 inch to $1\frac{1}{2}$ inch greater in diameter. If the stick is curved, it is so placed that the curvature is in a fore-and-aft plane, with the convex side forward. As to its diameter and taper, see Articles 24 and 26 of this Division. It is at first trimmed square; and for a depth below the hounds equal to from $\frac{1}{4}$ to $\frac{1}{5}$ of the length of the mast-head, it is left square. The head is made square, with the corners rounded off. Below the square part, the mast is made round, by first trimming off the corners, so as to make it "eight-square," or octagonal; then trimming off the corners of the octagon so as to make it "sixteen-square," and so on till it is sensibly round. The round part is "hanced into" the square part; that is to say, the one form passes gradually into the other by a fair curved surface. In a cutter's or schooner's mast the square part below the hounds is only one-fifth of the length of the masthead.

Single-tree masts are hooped with about five hoops at the head, and from one to three at the heel.

The *hounds-pieces* are coaked and bolted with five bolts on to the two sides of the mast just below the head; their length, in a vertical direction, is four-fifths of the length of the mast-head; their breadth, in a fore-and-aft direction, is equal to the fore-and-aft diameter of the mast added to that of the top-mast; the latter part of the breadth forms a pair of projections forward, called the *knees*.

Intermediate between single-tree masts and made masts are *cheeked single-tree masts*, used when the stick will not work to a sufficient breadth at the hounds. The *cheeks* are two pieces of timber coaked, bolted, and hooped to the mast at each side, in order to make up the required breadth in its upper part; and they extend from the cap down to midway between the hounds and the deck. The butt-ends of the cheek-pieces are usually placed upwards. The interval between the coaks, and that between the hoops, are nearly equal to the diameter of the mast. The hounds are made by cutting into the cheek-pieces

Fig. 1.



to the proper thickness; and the knees are separate pieces, bolted and notched on to the fore sides of the cheeks.

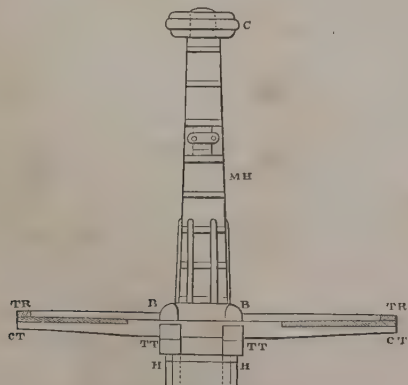
Made masts are built of several sticks, in thickness as well as in breadth. The number, shape, and arrangement of those pieces are varied in an indefinite number of ways, according to the size and shape of the mast required, and of the sticks available for making it; but the following general principles appear to be followed in almost every case:—

The pieces are coaked and bolted together at their faying surfaces, and the mast hooped in the intervals between the coaks; the length of such intervals being about equal to the diameter of the mast.

A piece of the mast, if necessary, may consist of two or three lengths, scarfed together, the length

of the scarf being between two and three times the diameter of the mast, so that three hoops may pass round it; and in such cases the scarfs should break joint with each other.

Fig. 2.



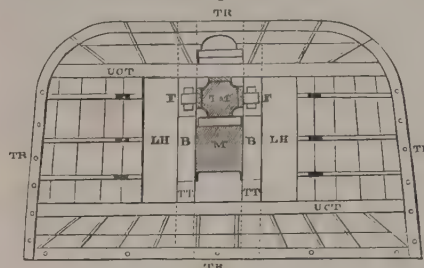
At the head of a ship's lower mast is a framing, of which Fig. 1 is a side or fore-and-aft view, Fig. 2 a thwartsip view from aft, and Fig. 3 a plan. In these figures, M is the mast, MH its head, with a cleat at the aft side to support the slings of the lower yard; H the hounds, K the knees of the hounds.

TT are the *trestle-trees*, being two pieces of strong hard

timber lying fore-and-aft, and supported by the hounds and knees. Half the breadth of each trestle-tree rests on the hounds-piece below; the other half projects. The trestle-trees are slightly tapered toward the ends.

CT are the *cross-trees*, lying athwartships, and supported by the trestle-trees, upon which they are scored down; two-thirds

Fig. 3.



of the score being taken out of the trestle-trees, and one-third out of the cross-trees.

Between the fore cross-tree and the mast is a square hole large enough to admit of the topmast sliding freely through it.

B are the *bolsters*, to guard the upper sides of the trestle-trees from being worn by the rigging. They are covered with hides or canvas.

Upon the cross-trees and trestle-trees is supported the *top*, being a platform of a shape more or less like that shown in plan in Fig. 3; that is, having a curved front, and being somewhat narrower at the fore than at the after side, in order that it may interfere as little as possible with the sharp bracing of the top-sail. In Figs. 1 and 2 the top is shown in section. The *flat of the top* consists of two sets of planks lying respectively athwartships and fore-and-aft, and halved into each other where they cross; they are also bound together edgewise by horizontal bolts. On the upper side of the flat of the top is the *top-rim*, TR; and the *sleepers* or *upper cross-trees*, UCT, directly above and of the same breadth with the lower cross-trees, to which they are bolted, and about half the depth. In the centre of the top is a square hole; the two parts of this hole at the sides of the mast, between the trestle-trees and the flat of the top, are each called *lubber's hole* (LH in Fig. 3); and sometimes each lubber's hole is capable of being enlarged when required by lifting a hinged scuttle.

At the centre of the fore part of the top, in Fig. 3, is seen a semicircular hole, with its after edge guarded by a wooden or iron bolster, for the passage of the slings of the lower yard.

The side parts of the top-rim have iron plates bolted above and below them, called *futtock-plates*, and have oblong holes in them, from three to five in number at each side, for securing the upper ends of the *futtock-shrouds*. The after part of the top-rim has four or five sockets in it, to receive the lower ends of the stanchions of the *top-rail*. (The top-rail is not shown in the figure.)

CN, CN (Fig. 1), are two *chain-necklaces*, surrounding the lower mast just below the lower ends of the hounds-pieces, and serving to secure the lower ends of the futtock-shrouds (to be again mentioned in a later Chapter). RP is the *rubbing-paunch*, a piece of wood nailed on the fore side of the mast, to prevent yards or other spars, which are being raised or lowered, from being injured by the hoops.

C, in Figs. 1 and 2, is the *cap*, being a flat oval block of wood, hooped with iron (and sometimes of iron or of copper), which has in it two holes; the after hole is square, and fits upon a tenon on the top of the lower masthead; the forward hole is round, and of such a size that the *topmast*, T M, can slide easily through it. In the heel of the topmast is a square hole, called the *fid-hole*, through which passes a square pin or key of iron, called the *fid*, F; the two ends of the fid are supported by the trestle-trees, upon iron plates, called the *fid-plates*; and thus the weight of the topmast is borne.

The lower part of the lower masthead, just above the bolsters, B, is protected from the friction of the rigging by upright battens, shown in Figs. 1 and 2.

The details of the construction of tops vary very much in different vessels. Sometimes, instead of close platforms of plank-ing, they have open gratings.

SUMMARY OF PROPORTIONS.

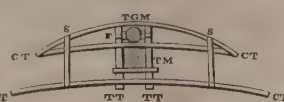
Hounds-pieces: height = height of masthead ×	about 0.8
" breadth, fore-and-aft, including knees = diameter of lower mast + diameter of topmast.	
" thickness = hounded length of topmast × $\frac{1}{15}$	
Mast-hoops: breadth, from 3 inches to 5 inches; thickness, from $\frac{1}{4}$ inch to $\frac{1}{2}$ inch.	
" drift, from $\frac{1}{16}$ to $\frac{1}{8}$ of diameter of mast.	
Coaks: diameter = diameter of mast ×	from $\frac{1}{4}$ to $\frac{1}{2}$
Bolts: diameter = diameter of mast ×	from .03 to .05
Trestle-trees: length = hounded length of topmast × 0.22	
" depth = length ×	about $\frac{1}{3}$
" breadth = depth ×	" $\frac{2}{3}$
Cross-trees: length = hounded length of topmast ×3
" breadth = breadth of trestle-trees.	
" depth = breadth ×	" $\frac{2}{3}$
Bolts for trestle-trees: diameter = depth of trestle-tree × from $\frac{1}{15}$ to $\frac{1}{10}$	
" for cross-trees: diameter = breadth of cross-tree × " $\frac{1}{10}$ to $\frac{1}{8}$	
Flat of top: thickness, from 2 inches to 3 inches.	
Top-rim: thickness = thickness of top ×	about $\frac{2}{3}$
Lubber's hole: length = length of trestle-trees ×	" 0.4
Futtock-plates: breadth, from $2\frac{1}{2}$ inches to 4 inches.	
" thickness, from $\frac{3}{8}$ inch to $\frac{1}{2}$ inch.	
Top-rail: about 3 feet high; stanchions and rail about 1 inch diameter.	
Cap of lower mast: length = $1\frac{1}{2}$ × diameter of lower masthead + 2 × diameter of topmast.	
" breadth = 2 × diameter of topmast.	
" thickness = $\frac{3}{8}$ × diameter of topmast.	
" square hole to fit tenon on lower masthead, tapered 1 in 12; round hole $\frac{3}{4}$ inch larger in diameter than topmast—viz., $\frac{1}{2}$ inch for leather, and $\frac{1}{8}$ for freedom.	
" Iron hoop, $\frac{1}{3}$ of depth of cap; thickness, $\frac{1}{2}$ in. to $\frac{3}{8}$ in.	
Fid: length = given diameter of lower mast × $1\frac{1}{2}$.	
" depth = diameter of topmast × $\frac{1}{3}$.	
" breadth = $\frac{2}{3}$ depth.	
Fid-plates: 1 inch thick.	

30. *Topmasts*.—The position and mode of support of topmasts have been stated in the preceding Article. The fid-hole is lined with iron on the upper side. There is also commonly a sheave-hole near the lower end of the topmast, containing a sheave for the *top-rope*, or rope by means of which the topmast is raised and lowered. If necessary, the heel of the topmast is made to fit the hole between the trestle-trees, by nailing on filling-pieces, three of which are seen in Fig. 3.

The framework at the head of a topmast is illustrated by Fig. 4; in which T M is the topmast-head. The hounds-pieces are of elm or other wood of similar quality; their length is half of that of the topmast-head, and they are of such a thickness as to be able to pass through the hole in the cap of the lower mast. Each of them is secured to the topmast by coaks and bolts. Above the hounds are the *trestle-trees*, T T, and

above the trestle-trees the *cross-trees*, C T, usually three in number; the foremost cross-tree forms a convex curve, and is secured to the middle cross-tree at the ends; and between them is a space, over the centre of which the heel of the topgallant-mast, T G M, is supported by means of a fid, F. The cross-trees are connected together by means of two or more iron straps, S. Close abaft the topmast is seen a *short cross-tree*. Sometimes, instead of the foremost long cross-tree, there is another short cross-tree. The upper sides of the trestle-trees are guarded by bolsters. The *cap* of the topmast is similar to that of the lower mast.

Fig. 4.



SUMMARY OF PROPORTIONS.

Trestle-trees: length = hounded length of topgallantmast × about 0.22	
" depth = length ×	" $\frac{1}{3}$
" breadth = depth ×	" 0.56
Cross-trees: length of long cross-tree before the topmast =	
length of trestle-tree ×	2
length of long cross-tree abaft the topmast =	
length of trestle-tree ×	2.1 nearly.
length of short cross-trees = width over trestle-trees + 2 × breadth of trestle-tree.	
breadth = breadth of trestle-tree.	
greatest depth = breadth ×	about $\frac{2}{3}$

Long cross-trees are of a parallel depth for the middle third of their length, and taper on the under side to half that depth at their ends. Cross-trees are let down $\frac{2}{3}$ of their depth into trestle-trees; $\frac{2}{3}$ of score taken out of trestle-trees, and $\frac{1}{3}$ out of cross-trees. Diameter of bolts about $\frac{1}{12}$ of depth of trestle-tree.

Sometimes the spaces between the trestle-trees and under the cross-trees, close afore and abaft the topmast, are filled with chocks; and below those chocks and the trestle-trees is bolted or screwed a plate of iron, with a square hole for the mast.

Topmasts, if necessary, may be made or built of more than one stick, on the same principles with lower masts, but with fewer pieces.

31. *Topgallant and Royal Masts* are usually in one piece, and are round, except at the heel of the topgallantmast, which is eight-square. The fid-hole is in depth half the diameter of the topgallantmast, and in breadth two-thirds of its depth. Above it is a thwartship sheave-hole for the topgallant-rope, by means of which the mast is raised and lowered.

Just above the *stops* or *hounds* of the topgallantmast, is a copper *funnel*, or short hollow cylinder, fitting easily upon the pole above the stops, and having a projecting rim round its lower edge. The topgallant-rigging is fitted on this funnel; and when the topgallantmast is lowered or *struck*, the funnel, with the rigging attached, rests by means of its rim upon the cap of the topmast-head, and allows the pole to be lowered through it.

Just below the topgallant-stops, and just below the *royal stops* (above which the royal stays and backstays are attached), there are fore-and-aft sheave-holes.

When there is a separate or *fded* royal mast, the head of the topgallantmast is fitted up like that of a topmast.

32. *Bowsprit and Fib-boom*.—A bowsprit, like a mast, may be of a single tree, or made of more than one tree. It is round throughout its length, except for one-ninth of its length at the outboard end, which is four-square on the top and sides, and rounded below. This part is called the *head* of the bowsprit,

and has bolted to its sides two pieces of hard wood, square, or nearly so, in section, and of half the dimensions of the bowsprit. These are called the *bee-blocks*, and have sheaves in them for certain ropes. At intervals along the outboard part of the bowsprit, square projections are left for the securing of certain other parts of the rigging. Beyond the extremity of the head projects a square tenon for the *cap*. The bowsprit cap is like that of a mast, except that it often stands vertically, and is traversed obliquely by its two holes. The lower hole is square, to fit tightly on the tenon of the bowsprit; and the upper cylindrical, for the jib-boom to slide through easily, with $\frac{3}{8}$ inch of play. The obliquity of the axis of the latter hole, to a line perpendicular to the fore-and-aft sides of the cap, is the same with the steeve of the bowsprit. The principal dimensions of the bowsprit-cap are—

Length = diameter of jib-boom	×	about 5
Breadth = " "	×	2
Thickness = " "	×	$\frac{5}{8}$

At a distance of one-third of the length of the jib-boom, inward from the outer side of the bowsprit-cap, is fixed, on the upper side of the bowsprit, an iron *saddle*, for the heel or inner end of the jib-boom. Near the inner end of the jib-boom is a thwartship sheave-hole, for the *heel-chain* by which it is held out.

The *flying jib-boom* is sometimes in one piece with the jib-boom; and the jib-boom has a copper funnel for its rigging, like a topgallantmast. When separate, the jib-boom usually carries the flying jib-boom by means of a ring, called a *boom-iron*, projecting obliquely upwards and sideways from the upper part of the starboard side of the outer end of the jib-boom. The heel, or inner end of the flying jib-boom, either steps in a notch in the bowsprit-cap, or fits in an inner boom-iron.

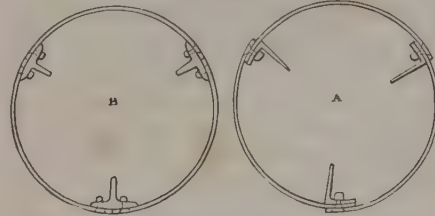
A *cutter's bowsprit* is made to *reef*, or run in, so as to shorten the outboard part in stormy weather. The inboard part and the reefing part are made square, with the angles slightly rounded; the rest of the outboard part is made round. In the same manner, the bowsprits of ships of war with ram bows are now fitted to reef.

33. *Fore-and-aft-rigged masts*, as those of cutters and schooners, and the mainmasts of brigantines and mizemasts of barques, have the hounds-pieces only one-fifth of the length of the masthead, and are sometimes fitted at the head with trestle-trees and cross-trees like those of a ship's topmast. Often, however, instead of a pair of trestle-trees, there is a *lower cap*, resting on the hounds, and similar to the cap on the head of the mast, except that the dimensions of the after part are somewhat greater, in order that the square hole may fit the masthead close above the hounds. The round hole in the fore part of the lower cap receives the heel of the topmast, which is supported there by means of a fid. Immediately before and abaft the mast, a pair of long cross-trees, like those of a ship's topmast, are supported by the lower cap, to which they are bolted. The upper surfaces of the side parts of the lower cap are protected against the friction of the rigging by means of bolsters, like those of a ship's trestle-trees.

34. *Iron and Steel Masts and Spars* are of the same outside figure and dimensions, or nearly so, with the wooden masts which they replace, but are hollow; the thickness of the shell being regulated by the principles stated in Article 27. The

seams or longitudinal joints of the shell are usually single-rivetted; the butts double-rivetted, except at the wedging-deck, where they are treble-rivetted; and the butts should break joint. Each breadth of plate in the circumference of the mast is usually stiffened by means of a longitudinal rib of L-iron or T-iron

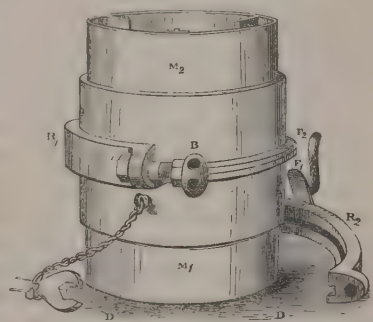
Fig. 5.



inside (see Fig. 5). These ribs in general run along the seams of the masts. If the seams are lap-jointed, L-shaped ribs may be used, as at A; if the seams are flush-jointed, as at B, T-shaped ribs are more suitable, placed so as to act both as stiffening ribs and as covering strips. Still further to stiffen the mast, the flanges of the ribs are sometimes connected together transversely by braces.

It is sometimes necessary for the safety of a ship during a storm to let the masts go overboard. A *parting-joint* for iron

Fig. 6.



or steel masts, contrived for that purpose by Messrs. Finch & Heath, is represented in Fig. 6. D, D is the weather-deck. M₁ is the lower, and M₂ the upper, division of a tubular iron or steel lower mast. Each of those divisions is strengthened at their junction by a collar, provided with a flange. F₁ is the flange of the lower division, and F₂ that of the upper division; and those flanges are accurately fitted to each other, so that they may be firmly and steadily clasped together by means of an internally-grooved ring, of which one half, R₁, is represented in position, and the other half, R₂, as lying on the deck. When both halves of the ring are in position, being held together by the screw-bolt, B, the parting-joint is at least as strong as any other part of the mast. So soon as the bolt is unscrewed, the ring falls off, and the mast goes overboard.

34A. *Tripod Masts* (the invention of Captain Coles, R.N.) are made of iron or steel, and consist of three equal, or nearly equal, diverging legs; one stepping on the keelson amidships, and answering the ordinary purposes of a lower mast, and the other two spreading at equal angles abaft of the first leg, and stepping on the floor. All three legs are so fastened to the ship's bottom and to the decks which they traverse, as to resist tension as well as thrust; and thus standing rigging is rendered unnecessary. The principal ropes of the running rigging required in action (viz., lower and topsail braces, and topsail tyes), are led down to between decks through the tubular legs, so as to be worked by men under cover, and to be themselves in a great measure protected from shot.

35. *Yards*, when of wood, may be either *single-tree* or *made*, according to their size, and the size of the sticks available for them. Iron and steel yards are like masts, thin hollow tubes stiffened by longitudinal ribs inside.

Lower yards are usually made eight-square for the middle half of their length, and topsail yards for the middle fourth part. Upon the upper and lower, and forward and aft faces of the eight-square part, are nailed hardwood *battens*, of a thickness equal to about one-eighth of the diameter of the yard, to protect the *bunt*, or middle part of the yard, from being chafed.

Wooden yards are strengthened by being hooped; the interval between the hoops over the battens is about twice the diameter of the yard, and the spaces between the battens and under the hoops are filled with chocks or filling pieces. On other parts of the yard, the distribution of the hoops is regulated mainly by the knots in the stick; but on lower and topsail yards, two of the hoops are so placed as to support the *inner boom-irons*, or *quarter-irons*, which are rings for supporting the inner ends of the studding-sail booms. The usual station for the quarter-irons is $\frac{1}{8}$ of the length of the yard from the outer end. The *outer boom-irons*, or *yardarm-irons*, are carried by necks projecting from straps or sockets at the yard-arm ends.

Along the upper side of a yard runs the *jackstay*, to which the sail is bent; being usually a rib of hard wood, with a series of oblong holes at its lower edge; or an iron rod passing through a row of eye-bolts; or for iron or steel yards, an L-shaped or T-shaped flange, with holes in it.

Made Yards may be constructed in various ways. For example, two trees, each of two-thirds of the required length, may be scarfed together butt-end to butt-end, the plane of the scarf being vertical, and its length the middle third of the length of the yard; or two trees, each of one-half of the length of the yard, may be placed end to end, and connected together by means of a pair of fish-pieces—the length of the fish-pieces being the middle half of the length of the yard. The scarf or fish-joints, as the case may be, are to be secured by coaks of about one-third of the whole diameter, and by hoops, at intervals of from once and a half to twice the diameter.

The slings, or middle part of a lower yard, is usually connected with and held near to the mast by means of a *truss*, consisting of a clasp-hoop round the mast, connected by an universal-jointed link with a hoop or a pair of hoops round the yard. In the case of upper yards, the same connection is made by means of a horse-shoe-shaped strap of rope or iron, called a *parral*, which fits loosely round the mast, so as to slide up and down when the yard is raised and lowered.

The following statement of dimensions for the principal iron fittings on yards is condensed from Fincham's table:—

Hoops on yards: breadth from 3 inches to 4 inches; thickness $\frac{1}{4}$ inch to $\frac{1}{2}$ inch.

Yard-arm irons: shank or neck, diameter $1\frac{1}{4}$ in. to 3 in.

“ strap; breadth $2\frac{1}{2}$ in. to 5 in.; thickness $\frac{1}{2}$ in. to $\frac{3}{8}$ in.

“ ring; breadth $1\frac{1}{2}$ in. to $3\frac{3}{4}$ in.; thickness $\frac{1}{4}$ in. to $\frac{3}{8}$ in.

“ bolts in straps; diameter $\frac{3}{8}$ in. to $\frac{1}{2}$ in.

“ hoops on straps; breadth $1\frac{1}{2}$ in. to $2\frac{1}{2}$ in.; thickness $\frac{3}{16}$ in. to $\frac{1}{2}$ in.

Quarter-irons, clasp and ring: breadth 2 in. to $4\frac{1}{2}$ in.; thickness $\frac{3}{8}$ in. to $\frac{1}{2}$ in.

Ferrules on ends of yards that have no boom-irons: breadth $1\frac{1}{2}$ in. to $2\frac{1}{2}$ in.; thickness $\frac{3}{8}$ in. to $\frac{1}{2}$ in.

Eyes on the ferrules: diameter from $\frac{3}{8}$ in. to 1 in.

36. *Booms and Gaffs*.—Upper studding-sail booms, and fixed

booms or outriggers, are carried by boom-irons. Lower studding-sail or swing-booms, and sometimes also the booms and gaffs of gaff-sails, have at the inner end a *goose-neck*, or iron shank with an eye on the end of it, which is shackled to a fixed eye-bolt, so as to make a sort of universal joint. The more usual fitting, however, for the inner end of a gaff, or of the boom of a driver or other gaff-sail, consists of *jaws*, forming a semicircle about one inch greater in diameter than the mast. The jaws are commonly of hard and tough wood, and are scarfed, and fastened with three or four hoops, on to the two sides of a tapering tongue formed at the inner end of the boom or gaff. The length of the scarf is about three times the diameter of the mast.

36A. A *trysail-mast* is usually from one-third to one-half of the diameter of the lower mast to which it belongs. It is of uniform diameter, and round from head to heel. The head is secured with a fid or bolt above the trestle trees; the heel steps either on the partners, or on a step, or on an eye carried by a clasp-hoop fastened round the lower mast.

37. A *Rolling Spar* is a spar upon which a sail is rolled for the purpose of reefing or furling it. This invention, the most useful of recent improvements in rigging, is due to Captain Cunningham, R.N. It has been applied chiefly to topsails; but it has been proposed to apply it also to other sails.

In one method of reefing and furling topsails (that of Mr. Cunningham), the topsail-yard itself is the rolling-spar. The yard-arms turn easily in two hoops which hang by the ropes called *lifts*; and the slings or centre of the yard forms a sheave, which is slung in the bight of a chain, whose two parts pass over two sheaves that revolve either in blocks hung from the topmast trestle-trees, or in sheave-holes in the topmast. By hauling on one or other part of that chain, the yard is made to revolve in either direction as required. In general, the forward part of the chain is made fast; so that lowering the after part of the chain at once lowers the yard and rolls up the sail; and hauling up the after part of the chain at once hoists the yard and unrolls the sail; a sort of action of the chain upon the yard known as *parbuckling*. The aperture required in the middle of the sail to allow the chain to work, and which divides the sail into two parts, is closed by a cloth called the *bonnet*, which is carried by travellers, and self-acting. The quarters of the yard are brought to an uniform outside diameter by properly shaping the battens. A small spar, in one or two lengths, of about one-third of the diameter of the yard, called the *chafing-spar*, hangs parallel to and abaft the yard, being connected with the parral and the lift-hoops; it serves to carry the boom-irons, and some blocks and other furniture that could not be properly carried by the rolling spar.

According to another method (that of Messrs. Colling & Pinkney), the rolling spar is distinct from the yard. It is supported directly in front of the yard, at its ends, by two journals, which turn in eyes at the ends of short arms projecting from hoops clasped round the yard; and also at about one-eighth of its length on each side of its middle point, by a pair of *crutches*, each of which is like a hoop with rollers on its inner surface, large enough to inclose the rolling spar with the whole sail rolled upon it, and having a gap or opening in front to allow the sail to pass through. Thus the sail is rolled and unrolled all in one piece. The rolling spar is slightly swelled in the middle; it has four battens upon its surface, one of which is of hard-wood and serves

for a jackstay; and it is made to revolve by the parbuckling action of two chains passing round its ends or arms.

37A. *Half-yards* are an invention of Mr. Cunningham, which have been tried on a small scale. Each sail resembles a square-sail; but its yard consists of two equal parts jointed in the middle, so as to be capable of swinging independently to any required

angle with a fore-and-aft plane. The two halves of the sail are bent to rings running on the two halves of the yard, and are set and furled independently by hauling them out and in. When the lee-half of a sail is set alone, it acts as a fore-and-aft sail; when both halves are set, as a square-sail. (See Transactions of the Institution of Naval Architects, 1862.)

CHAPTER III.

OF RIGGING AND SAILS.

SECTION I.—STANDING RIGGING.

38. *Channels* are flat ledges of wood or iron projecting outboard from the ship's sides, for spreading the shrouds or standing rigging at each side of the mast. The straining force to which they are subjected is principally compressive, and exerted inwards, and is represented by the arrow, V, in Fig. 6 of Chapter II., Third Division, Article 71, page 155; and to provide suitable means of resistance to that straining force, the channels should be on a level with the upper-deck beams. The *extent of projection* of the channels is made sufficient to carry the lower shrouds some five inches clear of the hammock-rails, either outboard or inboard. Formerly the shrouds always passed outboard of the hammock-rails; now they are often made to pass inboard of them, and are housed in the hammock berthing.

For an example of the position of the channels, see H.M.S. *Warrior*, Plates $\frac{B}{1}$, $\frac{B}{7}$.

Ships are often made without channels, the chain-plates being secured to the gunwale, or to the sheer-strake. For an example of this, see the *Formby*, Plate $\frac{F}{4}$.

The foremost end of the channels is usually so placed as to be nearly abreast of the fore side of the lower mast to which they belong. The foremost shrouds cross the channels nearly abreast of or a little abaft of the after side of the same mast.

The *length* of the channels, in a fore-and-aft direction, is on an average about one-half of the length above deck of the lower mast to which they belong, *including its head*—in other words, one-half of its length from upper deck to cap. About one-half of the length of the mast *from deck to hounds* is the length of the part of the channels occupied by the spread of the lower shrouds.

The thickness of the channels is about once and a half that of the skin of the ship's side where they are fastened on, supposing the material to be the same. Wooden channels are made from one-third to one-fourth thinner than this at the outer edge, which is sometimes bound with an iron bar or plate of the same depth, called the *guard-plate*. Channels are bolted to the ship's side edgewise with thwartship bolts at intervals of about three feet, and are supported from below by wrought-iron knees or brackets at about the same interval apart. The planks of which wooden channels are built are coaked together at their edges with coaks at about the same interval apart also.

39. The *Materials for Standing Rigging* are chiefly hempen ropes, coir ropes, iron-wire ropes, and iron chains.

Hempen ropes are classed according to the number and arrange-

ment of their strands: the following are the kinds chiefly used in rigging (see Division IV., Article 72):—

Hawser-laid rope = 3 strands.
Cable-laid rope = 3 hawser-laid ropes = 9 strands.
Shroud-laid rope = core or heart surrounded by 4 strands.

The *sizes* of rope are described by stating the girth in inches.

Coir ropes are made of cocoa nut fibres, and are useful where great lightness is required, because they float in water.

Wire rope consists generally of six strands laid or spun round a hempen core, each strand consisting of six wires laid the contrary way round a smaller hempen core. The spinning mechanism is so contrived that neither the wires nor the strands are twisted.

There are also wire ropes of three strands, each strand consisting of three wires. The following are approximately the sectional areas of metal contained in the several sorts of wire strands and ropes just mentioned:—

Area = Girth² ×

Six wires and hemp core,.....	0-056.
Six strands each of six wires,.....	0-038.
Three wires,.....	0-059.
Three strands each of three wires,.....	0-052.

Rigging Chain is usually of the unstudded or open-linked kind, with oval links. Its size is described by stating the diameter of the bolts of which the links are made.

The outside breadth of the links of a chain is about $3\frac{1}{2}$ times the diameter of the bolts of which it is made.

On account of the shocks and irregular strains to which rigging is exposed, a large factor of safety is allowed, the proof strength being *four times* the working load.

The following are the ordinary rules for calculating the proof strength and weights of ropes and chains, in *tons*; the dimensions being in inches:—

Rope or Chain.	Dimensions.	Multiplier for Proof Strength.	Multiplier for weight of 100 Fathoms.	Proof Strength in fathoms of Rope or Chain.
HEMPEN ROPE:—				
Hawser-laid,.....	Girth squared, ...	0-1875	0-0103	1820
Shroud-laid,.....	do. ...	0-15	0-01	1506
Cable-laid,.....	do. ...	0-12	0-0096	1250
WIRE ROPE (36 wires):—				
Iron,.....	do. ...	0-75	0-039	1923
Steel,*.....	do. ...	1-125	0-04	2812
RIGGING CHAIN,....	{ Diameter of } { bolt squared, }	12-00	2-9	414

* The estimates of the comparative strength of iron and steel wire ropes here given, are founded upon some experiments by Messrs. Jones, Quiggin, & Co., on the ultimate tenacity of steel-wire ropes, and experiments by the Editor of this Treatise, on the ultimate tenacity of charcoal iron-wire ropes; from which it appears that steel ropes are about once and a half the strength of iron ropes of the same girth. The proof strength is taken in each case, agreeably to ordinary practice, at about three-sevenths of the mean breaking load.

The breaking load may be estimated as ranging, for iron, from 2 to $2\frac{1}{2}$ times, and for hemp, from 2 to 3 times, the proof loads given in the preceding table, which are taken as nearly as possible at half the *least* ultimate strength of good material.

The following table gives the comparative dimensions of chains and ropes of equal strength:—

Chain. Diameter of Bolt.	Wire Rope.		Hempen Rope.			
	Steel. Girth.	Iron. Girth.	Hawser-laid. Girth.	Shroud-laid. Girth.	Cable-laid. Girth.	
1	...	$3\frac{1}{4}$...	4	...	8
...	9
...	10

Wire and chain rigging is preserved by galvanizing, or coating with zinc. That process makes wire ropes somewhat softer and more extensible, but does not diminish their tenacity. Hempen rigging is preserved by means of tar. Both wire and hempen ropes for standing rigging are often protected against chafing by *worming*; that is, winding spun yarn round the rope, so as to fill the hollows between the strands; *parcelling*, that is, covering the rope with a narrow strip of tarred canvas wound spirally round it; and *serving*, or winding spun yarn round it against the lay in close coils, so as to cover it completely. Ropes are usually wormed and parcelled before being served.

Ropes are connected with each other, and with spars and blocks, by means of various sorts of splices, hitches, bends, knots, seizings, &c., the art of making which belongs properly to seamanship, and not to shipbuilding, and will therefore not be described here in detail; but the following general principles may be stated:—

I. A *short splice*, in which the strands of two ropes, or of two parts of the same rope, are interwoven for a length equal to from once to twice the girth of the rope, is nearly, if not quite, as strong as the original rope, if well made, whether in hemp or wire. A wire rope requires a somewhat longer splice than a hempen rope, because the friction is less.

II. A *long splice*, in which the strands of two ropes are interwoven for a length of from half a fathom to a fathom, with pieces of different lengths cut off the ends of those strands, so that the splice is of the same thickness with the original rope, may be estimated as equal in strength to the original rope, less one strand.

III. An *eye-splice*, in which the end of a rope is bent round into an eye, and spliced into the "standing part" of the same rope, if well made in hemp or in wire, is as strong as the original rope, or nearly so; and is the strongest way of securing a rope round a spar, block, dead-eye, thimble, &c. An eye made by *seizing* the end of a rope to the standing part (that is, placing them alongside of each other, and lashing them together) is not so strong. An *iron socket*, rivetted on the end of a wire rope, injures the rope, and makes a weak fastening.†

IV. *Knots* are not applicable to wire ropes. The principle of a secure knot for a hempen rope is, that no two parts of the rope, which would move in the same direction if the rope were to slip, should lie alongside of and touching each other. When applied to knots for joining two lengths of rope, this principle leads to the rule, that the *standing part of one rope, and the end of the other, should not lie side by side*; and this is what distinguishes the knots used by seamen (as the "sheet bend,"

* For information on those matters, see Darcy Lever's "Young Sea Officer's Sheet Anchor;" Nares "On Seamanship;" Biddlecombe's "Art of Rigging;" Kipping's "Rudimentary Treatise on Mast and Rigging;" Boyd's "Naval Cadet's Manual," &c.

† In a series of experiments made by the Editor of this Treatise on the tenacity of iron-wire ropes, an eye-splice round a dead-eye or thimble was found to be the only fastening which neither weakened the rope nor gave way before it. Rivetting into an iron socket was found to weaken the rope; and seizings always gave way to a load less than the breaking load of the rope.

"bowline knot," "carriek bend," "reef knot," &c.), from what they call "granny's knots" and "slippery hitches."

40. *General Description of Standing Rigging*.—Standing rigging consists mainly of ropes in two sorts of positions: in a fore-and-aft vertical plane, when they are called *stays*; and in an oblique position, extending downwards and backwards in pairs from the head of a mast, when they are called *shrouds* and *backstays*. The general nature of the straining action brought upon the masts and standing rigging, by the pressure of the wind on the sails, has already been shown in Fig. 6 of Chapter II. of the Third Division, Article 71, page 155; and that action always produces thrust on the mast, and tension on the standing rigging. When the sails are filled, and braced directly athwartships, the tension is equally divided between the shrouds and backstays at either side of the mast; when the sails are braced obliquely, the tension is greatest on the rigging at the weather side; and when the sails are braced up very sharp, the whole tension falls upon the weather rigging. When the sails are laid aback, the tension is thrown upon the stays, sometimes assisted more or less by the foremost of the shrouds at the weather side.

In practice, the moment of the pressure of the wind on the sails is usually resisted partly by the combination of direct thrust along the mast with tension along the rigging, as already described, and partly by the resistance of the mast to bending, exerted with maximum moment at the partners, where the mast is wedged into the deck. The latter kind of action involves much more severe stress on the material of the mast than the former; and accordingly, it is in general at the partners that masts are found to be "sprung," or overstrained.‡

The following is a description of the standing rigging of a full-rigged ship, as shown in Plate F. Vessels of less complex rig differ from this example only in having fewer parts in their rigging.

- 1, Bowsprit-shrouds.
- 2, Bob-stay.
- 3, Fore-stay, with fore-preventer-stay alongside of it: one to act if the other is carried away.

The term *preventer* is applied to any part of the rigging intended to act in the event of another part being lost or disabled.

- 4, Fore-topmast-stay (with preventer-stay).
- 5, Inner jib-stay.
- 6, Outer jib-stay.
- 7, Fore-topgallant-stay.
- 8, Flying jib-stay.
- 9, Fore-royal-stay.
- 10, Martingales and back-ropes.
- 11, Fore, 23, Main, 35, Mizzen shrouds.
- 12, " 24, " 36, " futtock-shrouds.
- 13, " 25, " 37, " topmast-shrouds.
- 14, " 26, " 38, " topgallant-shrouds.
- 15, " 27, " 39, 42, " topmast-backstays.
- 16, " 28, " 40, " topgallant-backstays.
- 17, " 29, " 41, " royal-backstays.
- 30, Main-skysail-pole backstays.

Crossing the shrouds are seen the *ratlines*, forming the ladders by which men go aloft.

- 18, Main-stay (with preventer-stay alongside of it).
- 19, Main-topmast-stay (with preventer-stay).
- 20, Main-topgallant-stay.
- 21, Main-royal-stay.
- 22, Main-skysail-pole stay.
- 31, Mizzen-stay.
- 32, Mizzen-topmast-stay.
- 33, Mizzen-topgallant-stay.
- 34, Mizzen-royal-stay.
- 35, Foot-ropes.

‡ Mr. Lamport, in a paper in the Transactions of the Institution of Naval Architects for 1863, recommends that masts should not be wedged, but left fitting easily in the partners; in order that they may be relieved from bending stress, and subjected to direct thrust only.

The following parts are not seen in the Plate:—

The *heel-chain*, for holding out the jib-boom, and the *crupper-chain*, for lashing it down to the bowsprit.

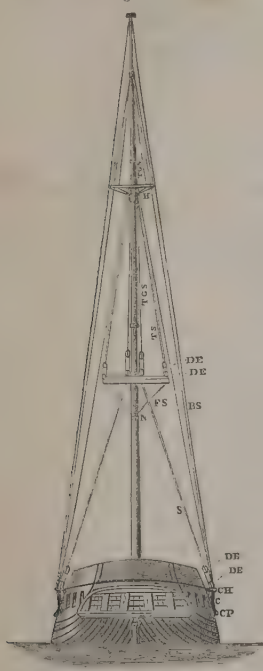
Gammonings, or chains for lashing the bowsprit to the knee of the head, through the holes shown in Division IV., Chapter II., Fig. 2, page 184.

In an iron or steel vessel, whose bowsprit is supported by a tube, or by a pair of rings framed to thwartship bulkheads, gammonings are unnecessary.

Jib-guys and *flying jib-guys*, from the jib-boom end and flying jib-boom end to the ends of the spritsail-yard (or spritsail-gaffs, as the case may be); and *after guys* and *jumpers*, from the ends of the spritsail-yard (or gaffs) to the ship's bows: these ropes act as shrouds to the jib-boom and flying jib-boom; also *spritsail-yard* or *gaff-topping-lifts*, from the ends of those spars to the bowsprit-cap.

Fore, main, and mizen mast-head pendants, being short ropes, with blocks at the end, hanging from the lower mast-heads:

Fig. 1.



usually four each to the foremast and mainmast, and two to the mizenmast. They are used in setting up masts and rigging.

Fore, main, and mizen topmast burton pendants, two to each mast, hanging from the topmast-heads, for similar purposes to the above.

Horses, or *ridge-ropes*, two in number, from the knight-heads to the upper part of the bowsprit-cap, for the safety of men walking out upon the bowsprit in rough weather.

41. *Fitting, Securing, and Setting up of Standing Rigging.*—The standing rigging of masts is usually fitted on the mast-head; that of lower masts immediately above the bolsters, where the mast-head is guarded by battens (see Article 29 of this Division); and that of topmasts, topgallant-masts, and royal-masts, upon the copper funnels, mentioned in

Articles 30 and 31. *Mast-head pendants*, and most of the *shrouds* and *backstays*, are put on in pairs, each pair being one rope; the bight of the rope forms a seized eye that fits over the mast-head, and the pair of shrouds or of backstays come down at the same side of the mast. The aftermost shroud of a mast at each side (called the *after swifter*) is usually an odd one, and has an eye-splice at the upper end fitting over the mast-head. When there is an odd backstay, it is usually the foremost one.

Stays are usually forked at the upper end, the two parts of the fork having eyes at their ends, which are lashed together at the after side of the mast-head.

The rigging of a mast is usually put on in the following order:—Mast-head pendants; shrouds, starboard and port alternately, commencing with the foremost pairs and ending with the after swiflers; backstays, commencing with the foremost; lower

and topmast stays; but topgallant and royal stays are put on before the shrouds and backstays.

The lower ends of the standing rigging are usually secured by means of a pair of *dead-eyes*, of *hearts*, or of *thimbles*, drawn together by means of three or four turns of a rope called a *lanyard*. A *dead-eye* is a round block of some very hard, tough, and durable wood (*lignum-vitæ* being almost always employed in the merchant navy, and elm in the Royal navy), or of cast iron,* with three or four holes in it for the lanyard, and a *score* or groove round its circumference for the shroud or backstay (several dead-eyes are shown in Plates $\frac{1}{1}$ and $\frac{2}{2}$). A *heart* differs from a dead-eye chiefly in having one large hole, with scores at the end for the turns of the lanyard. A *thimble* is an iron ring fitting into an eye at the end of a stay. In some cases, a *purchase*, consisting of two blocks with revolving sheaves drawn together by a lanyard, is used for setting-up a stay. Each row of upper dead-eyes are kept in their places by a *stretcher* or sheer-pole seized to the shrouds.

Sometimes, for setting-up iron rigging, a right and left handed screw coupling is used instead of a lanyard and dead-eyes.

ORDINARY NUMBERS OF SHROUDS, BACKSTAYS, AND BOBSTAYS.

Masts, each:	Backstays.	Shrouds.
Royal,	2	0
Topgallant,	2 to 4	4
Topmast,	4 to 6	6 to 10
Lower,	0	8 to 18
	Bobstays.	Bowsprit Shrouds.
BOWSPRIT,	3 to 1	4 to 2

Lower Shrouds are secured at the channels, as shown in Fig. 1; where S is a lower shroud; D E, D E, a pair of dead-eyes connected by a lanyard: one of the dead-eyes is "turned in," as it is called, at the lower end of the shroud; the other is secured at the upper end of one of the *chains*, C, being an iron rod or bar which passes through a notch in the edge of the channel, C H, and ends at the *chain-plate*, C P, where it is fastened to the ship's side with two bolts. The backstays, B S, are secured in the same way, but with smaller dead-eyes and chain-plates.

Each *topmast shroud*, T S, is set up at its lower end by means of a pair of dead-eyes and a lanyard; the lower dead-eye being secured to the upper end of a *futtock-shroud*, F S, being an iron rod which passes through a hole in the top-rim. The lower ends of the futtock-shrouds are fastened alternately to the upper and lower of the two chain-necklaces at N. The angle which the futtock-shrouds usually make with the axis of the mast is 45°.

The *topgallant-shrouds*, T G S, consisting of two pairs, reeve through holes in the *horns*, or ends, of the topmast cross-trees, and then round rollers in the *fairleader-hoop*, H; after which the lower ends of each pair of shrouds are spliced together, and the bight thus formed made to reeve through a thimble in the strop of a block, which is connected by a lanyard with a lower block, fastened to the eye of one of the lower shrouds. The object of this arrangement is to insure equal tension on the two shrouds of each pair.

The *fore-stays* are set up to collars on the bowsprit, or straps on the bow; the *main-stays* to the knight-heads, or to a cross-piece before the fore-bitts, or to a plate-bolt in the deck, each

* Cast iron dead-eyes were first used by Mr. J. R. Napier.

with a pair of hearts and four turns of a lanyard; the *mizen-stay* has usually a forked end, set up to a pair of eye-bolts in the deck.

The *fore-topmast-stays* are rove through holes in the bee-blocks of the bowsprit, and set up to the knight-heads with lanyards.

The *jib-stay* and *fore-topgallant-stay* are rove through holes in the jib-boom end, and the *flying-jib-stay* and *fore-royal-stay* through holes in the flying-jib-boom end; and then all four are rove through holes in the dolphin-striker, and set up to the knight-heads with lanyards.

The *main-topmast-stays* are set up with lanyards to bolts in the deck before the foremast, and sometimes in the knight-heads.

The *mizen-topmast-stay* is set up to the mainmast-head.

The *main* and *mizen topgallant* and *royal stays* are each usually rove through a sheave at the head sometimes of the next lower division, and sometimes of the next again, of the mast next before the mast to which they belong, and thence led down to the tops, where they are set up to the eyes of lower shrouds.

Gammonings are made of seven, nine, or eleven turns of chain, shackled at one end to the bowsprit. Each successive turn is taken outside the preceding turn over the *gammoning-fish* on the top of the bowsprit, and inside the preceding turn through the gammoning-hole in the knee of the head. Each turn is separately bowed taut with a purchase before passing the next turn, and stoppered by driving large nails through the links of the chain into the gammoning-fish; and the end of the chain is wound or *frapped* round all the parts of the gammoning from the knee of the head up to the bowsprit, so as to bind them together. There are usually two gammonings, and the outer one is put on first; because, as it has most leverage, it would, if it were put on last, cause the inner gammoning to become slack.

Bobstays and *bowsprit-shrouds* are very often of chain. They are shackled to bolts or holes in the cutwater and to bolts at the bows respectively, and are set up at their outer ends, being secured with a pair of hearts and a lanyard to collars on the bowsprit-head. In armed vessels, the bowsprit-shrouds are secured to the bolts in the bow with slip-hooks, in order that they may be let go when the bow-guns are to be fired.

The *jib-guys* and *flying-jib-guys* are fitted with eye-splices on the ends of the jib-boom and flying-jib-boom respectively, and are secured at their inner ends to the spritsail-yard or spritsail-gaffs, as the case may be. They are set up by setting taut the jumpers and after-guys, which lead from the spritsail yard or gaffs to the bows.

The *jib-martingale* is secured at its ends to the jib-boom end and to the dolphin-striker, and is set up by setting taut the *back-ropes* of the dolphin-striker, which lead to the bows. The *flying-jib-martingale* is rove through a hole in the dolphin-striker, and set up in the head of the ship.

The *spritsail-gaff topping-lifts* are rove through blocks on the bowsprit-cap, and set up to the knight-heads.

The *heel-chain* is in two pieces, a longer and a shorter, shackled to opposite sides of the bowsprit-cap. The longer piece passes round abaft the heel of the jib-boom, and is secured to the shorter piece with a chain-slip. The *crupper-chain* passes round the bowsprit and the heel of the jib-boom, and is secured at one side of the bowsprit with a chain-slip. (In the vessel represented in Plate *P*, there is no heel-chain nor crupper-chain; as the heel of the jib-boom is held down to the bowsprit with a hoop, and

abuts against a block that is bolted to the upper side of the bowsprit.)

As a general rule in the setting up of rigging, *the rope which acts with the greatest leverage on a spar should be set up taut first*; otherwise the setting of it up will slacken those which have been set up before it with less leverage.

42. *Fore-and-aft-rigged Masts*.—The masts of cutters and schooners have usually from four to eight *shrouds*, the most common number being six, a *stay*, and (to the foremast of schooners and mast of cutters) a *preventer-stay*. The *fore-stay* of a cutter, and the *fore-preventer-stay* of a schooner, are set up to the stem; the *fore-stay* of a schooner, to a collar on the bowsprit. The bowsprit has usually two shrouds and a bobstay; in a cutter, which has a running bowsprit, these are hauled in by purchases when the bowsprit is run in. The cutter's bowsprit is run out by a *heel-chain*. The *main-stay* of a schooner leads from the cap of the mainmast to the cap of the foremast, so as not to be in the way of the foresail-gaff. Sometimes the mainmast has a pair of *jumper-stays*, that is, moveable stays, leading from the head of the mainmast to a pair of eye-bolts in the deck close to the after-part of the fore-rigging, the weather jumper-stay alone being set up.

The *jib-stay* of a cutter leads from the mast-head to a traveller on the bowsprit; that of a schooner, from the head of the foremast to the jib-boom end, where it reeves through a sheave-hole, thence through the end of the dolphin-striker, and thence inboard.

The topmasts have usually four shrouds, fitted like the topgallant-shrouds of a ship (see Article 41), being rove through the cross-trees, and set up at the channels; a stay; and from two to four backstays.

The *topmast-stay* of a cutter reeves through a sheave-hole in the end of the bowsprit, and is thence led inboard; the *fore-topmast-stay* of a schooner reeves through the ends of the jib-boom and dolphin-striker, and is thence led inboard.

43. *Dimensions of Standing Rigging*.—To find the length of any part of the standing rigging, when its projections on a fore-and-aft and a thwartship rigging plan are given, Rule III. of Article 9 of the Second Division is to be used.

The following general rules as to the proportionate sizes of standing rigging are condensed from the tables given in the works of Biddlecombe and Kipping. When not otherwise specified, these proportions apply to the girths of hempen ropes, hawser-laid or shroud-laid; and the sizes of iron or steel wire ropes, or of chains, of equal strength, can be calculated by the aid of the table of proportions given in Article 39 of this Division:—

BOWSPRIT:—	About
Gammoning and bowsprit-shrouds—iron chain: diameter of bolts = diameter of bowsprit ×	3/4
Bobstays—iron chain: diameter of bolts = diameter of bowsprit ×	3/4
Lanyards—shroud-laid ropes: girth = diameter of chain-bolts, × 4.	
Horses or man-ropes = lanyards of bobstays.	
JIB-BOOM:—	
Stay and guys: girth = diameter of boom ×	0.40
Martingale stay = " "	0.50
" back-ropes = " "	0.35
Foot-ropes = " "	0.25
Heel-chain: diameter of iron = diameter of boom, ×	0.04
FLYING JIB-BOOM:—	
Stay and guys: girth = diameter of boom ×	0.35
Martingale = " "	0.40
Foot-ropes = " "	0.30
Heel-lashing = " "	0.25

LOWER MASTS, square-rigged:—		About
Shrouds and pendants: girth = diameter of mast ×	0.375	
Stays = shrouds ×	1.50	
Lanyards = "	0.50	
Ratlines = "	0.15	
TOPMASTS, square-rigged:—		
Shrouds: girth = diameter of mast ×	0.375	
Stays and backstays: " " ×	0.45	
Lanyards = shrouds ×	0.50	
Ratlines: "	0.15	
Pendants: diameter of mast ×	0.30	
Futtock-shrouds—single iron rods: diameter = diameter of mast ×	0.03	
TOPGALLANTMASTS:—		
Shrouds, stays, and backstays: girth = diameter of mast ×	0.50	
Lanyards = shrouds ×	0.50	
Royal-stay and backstays = topgallant-stay ×	0.50	
Lanyards of do. = royal-backstays ×	0.40	
SCHOONER'S BOWSPRIT:—		
Shrouds and bobstays—iron chain: diameter of bolts = diameter of bowsprit ×	0.05	
CUTTER'S BOWSPRIT:—		
Shrouds—iron wire: girth = girth of bowsprit ×	0.12	
Bobstay pendants—iron wire: girth = girth of bowsprit ×	0.16	
SCHOONER'S JIB-BOOM:—		
Jib-stay and guys: girth = diameter of boom ×	0.625	
Martingale-stay—chain: diameter of bolt = diameter of boom ×	0.06	
Martingale-back-ropes: girth = diameter of boom ×	0.56	
Foot-ropes = " " ×	0.30	
Heel-chain: diameter of iron = " " ×	0.05	
LOWER MASTS, fore-and-aft-rigged:—		
Shrouds, pendants, and schooner's jumper main-stays: girth = diameter of mast ×	0.45	
Stays: girth = diameter of mast ×	0.64	
Fore-storm-stay of schooners,	0.32	
TOPMASTS, fore-and-aft-rigged:—		
Shrouds, stays, and backstays: girth = diameter of mast ×	0.40	
Any lanyard: girth = girth of rope set up by it ×	$\frac{1}{2}$	
Length of rope for a shroud-eye = girth of mast-head ×	$1\frac{1}{2}$	
Diameter of a dead-eye = girth of hempen shroud ×	$1\frac{1}{2}$	
All hempen standing rigging to be stretched before being fitted until its length is increased to original length ×	$1\frac{1}{4}$	
Ratlines apart, from 15 to 16 inches.		

44. The *Standing Rigging* of a yard consists mainly of the jackstay, head-earing strops, parral and truss, slings, foot-ropes and stirrups, and Flemish horses.

Jackstays of wood and iron have already been mentioned in Article 35. When of hempen rope, the jackstay, in two lengths, runs along the upper side of the yard, through a series of eye-bolts at intervals of about twice the diameter of the yard; and is set up taut by means of lanyards at the slings of the yard. At the outer ends of the jackstay are the *head-earing strops*, for bending the upper corners of the sail to.

Parrals and *trusses* have been mentioned in Article 35.

The *slings* of a yard consist of a double strop of rope or chain, passing round the bunt, or middle of the length of the yard. In lower yards, the slings are always of chain, and form a bight at the upper end which passes over a chock at the after side of the lower mast-head.

Foot-ropes for the men to stand upon are shown in Plate $\frac{F}{4}$, and marked 55. They are hung from the yards by means of *stirrups*. *Flemish horses* are short separate foot-ropes for the yard-arms.

ORDINARY DIMENSIONS:—		About
Jackstay: (hempen) girth = diameter of yard ×	0.25	
Slings: " "	0.42	
Parral: " "	0.33	
Foot-ropes: " "	0.30	
Stirrups: " "	0.25	

Dimensions for wire-ropes and for chains of equal strength may be computed by the aid of the proportions given in Article 39.

SECTION II.—SAILS.*

45. *Materials of Sails.*†—A sail is made up of strips of canvas, called *cloths*. These are from 18 to 24 inches broad, 24 inches being the most common width; and of various thicknesses, usually numbered from 0 to 8, No. 0 being the thickest. A *bolt* of canvas is from 39 to 40 yards long. The best canvas is made of long-fibred flax of the strongest quality; British and Irish flax being preferred. Inferior qualities are made of tow or short-fibred flax, hemp, and cotton. The yarns are carefully washed and boiled before being woven, to prevent mildew; and for the same reason no dressing is used. The following Table shows the weight and strength of British Royal Navy canvas.

Number of Canvas.	Length of a Bolt.	Weight of a Bolt 24 inches wide.	Tenacity, by Testing Machine. (see Note, p. 246.)		Use.
			Wett. lbs.	Warp. lbs.	
0 ... 39	...	48	Courses, lower staysails, trysails.
1 ... 39	...	46	480	340	Courses, lower stay-sails, trysails, awnings.
2 ... 39	...	43	460	320	Courses, topsails, lower stay-sails, try-sails, spankers, awnings.
3 ... 39	...	40	440	300	Courses, topsails, spankers, jibs, lower and topmast stay-sails.
4 ... 39	...	36	400	280	Topsails, topgallant-sails, spankers, jibs, topmast stay-sails.
5 ... 39	...	33	370	260	Topsails, lower and topmast studding-sails, spankers, jibs, upper stay-sails, gaff-topsails.
6 ... 39	...	30	350	250	Topgallant-sails, studding-sails, jibs, flying jibs, upper stay-sails, gaff-topsails, cutters' and schooners' cross-jack-sails and square topsails, sails of boats.
7 ... 40	...	27	390	330	Topgallant-sails, studding-sails, flying jibs, royal stay-sails, cutters' and schooners' topsails, sails of boats.
8 ... 40	...	25	380	310	Royals, skysails, topgallant and royal studding-sails, cutters' and schooners' topgallant-sails, save-alls, sails of boats.

The second quality of canvas, called "Merchant Navy," is fully one-third weaker than Royal Navy canvas of the same weight. Cotton canvas is from $\frac{1}{10}$ to $\frac{3}{4}$ of the strength of Royal Navy canvas of the same weight.

The durability of canvas, in store, depends upon its being kept clean, dry, and in pure air.

The linings, or doublings, of sails are made of canvas from one to three numbers lighter than the body of the sail.

The *twine* with which sails are sewed weighs at the rate of from 360 to 430 fathoms to the lb.; and on an average one lb. of twine is required to sew 160 yards of seam.

The average weight of a ship's sails is at the rate of from 2200 to 1800 yards of canvas, 24 inches wide, to the ton.

46. *Parts of a Sail.*—Some of the principal parts of a sail have been mentioned in Article 6 of this Division, as the head, foot, and leeches; the clews of a square-sail, and the clew and tack of a fore-and-aft sail; the peak and nock (or throat) of a trysail or spanker; &c.

The *bunt* of a square-sail means the middle part.

The *bolt-rope* is a rope sewed round the edges of a sail to

* For detailed information on the subject of sails, see "Sail-making as practised in the Royal Navy," and Mr. Kipping's "Elementary Treatise on Sails and Sail-making."

† The information here given respecting canvas, is for the most part extracted from a paper by Mr. Peter Carmichael, of Messrs. Baxter Brothers & Co, Dens Works, Dundee, which will probably be published in the "Transactions of the Institution of Engineers in Scotland, with which is incorporated the Scottish Shipbuilders' Association," for 1865-66.

strengthen it. At the head of the sail it is called the *head-rope*, at the leeches the *leech-ropes*, at the foot the *foot-rope*. At each of the lower corners or *clews* of a sail, a ring is made for attaching the *sheet*, either by making an eye on the bolt-rope, with or without an iron thimble, or by having separate eyes and thimbles on the leech-rope and foot-rope, connected with each other by means of an iron ring. A square-sail is roped on the after side, and a fore-and-aft sail on the port side. A sail is bent to a yard or gaff, with the roped side to the spar.

Sails are lined or doubled with an additional thickness of canvas called the *tabling*, on the roped side. On square-sails this extends all round inside the bolt-ropes; on fore-and-aft sails it usually extends only to the luff, the head, and the lower part of the weather leech. Courses and topsails have also a doubling of canvas on the after side at each of the *reef-bands* (Plate $\frac{V}{4}$, 53), at the *belly-band*, which runs horizontally midway between the lowest reef-band and the foot of the sail, and at the *bunt-line cloths*, from two to four in number, which run up and down from the belly-band to the foot of the sail. A topsail is moreover doubled at the middle of its lower part with a piece of canvas called the *top-lining*, to protect it against being chafed by the top; and near the middle of each leech with a piece called the *reef-tackle patch*.

Loops called *cringles* are worked into and round the bolt-ropes of sails. Those at the upper corners of a square-sail are called *head-cringles*, and have spliced or otherwise secured to them loops of rope called the *head-earings*, which are lashed to the head-earing strops on the yard-arms. The other cringles are for parts of the running rigging to be specified further on.

The head-rope of the sail is secured to the jackstay on the yard with *robands* passing through eyelet-holes; those are pieces of a sort of rope called *sennit*, made by plaiting yarns together; and they are in number from one and a half to two to each cloth of the sail.

In each reef-band there is a row of eyelet-holes, usually one to each cloth of the sail, through which are rove either a row of *reef-points*, or two *reef-lines*. A course has usually two reefs; a topsail four; a spanker, or a trysail, three.

Reef-points are like robands, and measure in length about twice round the yard. They hang half before and half abaft the sail, and are stitched by the middle to the upper edge of the hole in square-sails, and the lower edge in fore-and-aft sails. Reef-lines pass back and forward through the holes like a lacing; in reefing the sail, the forward parts of the reef-lines are tied to the after parts over the yard with a row of pieces of sennit called *reef-beckets*, which hang from the jackstay. In reefing by means of a rolling spar those parts are not wanted, with the exception of the close-reef points, to be used in shifting a split sail. When there are an upper and lower topsail (as in Plate $\frac{V}{4}$) the upper topsail is lowered before being reefed or furled so as to hang to leeward of the lower; and furling the upper topsail answers instead of close-reefing.

47. *Figures of sails*.—In addition to the statements in Chapter I. of this Division regarding the figures and dimensions of sails, the following explanations have to be made:—

The edges of sails which are bent to spars, such as the heads of square-sails and gaff-sails, the luffs of gaff-sails, and their feet when laced to their booms, are made straight. The leeches

of square-sails are usually straight; but those of topsails are sometimes slightly hollowed.

The *roach* of a sail is the concave curve to which the foot of it is sometimes cut. Courses are usually roached to a depth equal to about one-eighth of the depth of the sail, and are straight for the middle three-fifths of the foot. Upper square-sails, when they are not straight at the foot, are roached to such a depth as may be required in order to clear the stays that pass below them. Square-topsails of cutters and schooners are often very deeply roached, because of the cross-jack yards being far below the fore-stay, and in light winds the roach is filled up by means of a small sail called a *save-all*. All unnecessary roaching of sails is an evil, as it diminishes the area of canvas to no purpose. Spankers and other gaff-sails are sometimes cut to a convex curve at the foot, being an arc of a circle whose radius is nearly equal to the sum of the lengths of the fore and after leeches of the sail. Their after-leeches are often made slightly convex; and so also are the fore and after leeches and feet of jibs. The use of the slight convexity sometimes given to the edges of fore-and-aft sails appears to be to counteract the tendency to become hollow at the edges, which arises from the canvas undergoing greater tension near the corners of the sail than elsewhere; and in particular, the convexity given to the fore-leech of a jib prevents the tension of the jib-sheet from producing an inward deflection of the stay.

The old practice in sailmaking is, by suitably varying the breadths of the seams, to make the sail have a *belly* or *bag* towards the middle, in order to increase the concavity of the surface that it presents to the wind. This may slightly increase the efficiency of sails in running before the wind; but it greatly diminishes their efficiency when close-hauled; and the most efficient sails on the whole are those whose figures, when not strained by the pressure of the wind, are perfectly flat.

48. *Dimensions of Canvas for Sails*.—The dimensions of a sail, as determined by the principles stated in Chapter I. of this Division, and shown on a sail-draught, are those to which the sail is ultimately to come by stretching. The sail when newly made must have its dimensions less than the intended ultimate dimensions, by a fraction sufficient to allow for that stretching. That fraction may be estimated as ranging, for the head and leeches of a sail, from $\frac{1}{30}$ to $\frac{1}{20}$, and for the foot from $\frac{1}{15}$ to $\frac{1}{10}$.

In computing the number of cloths of canvas required in order to make a given width of sail, regard must be had to the breadth taken up by the seams. This may be done by making a deduction from the total breadth of a cloth of canvas, so as to leave an effective breadth, by which the width of sail required is to be divided, in order to find the number of cloths. For example, the total breadth of a cloth being 24 inches, the following may be taken as the effective breadths for different classes of sails (according to Mr. Kipping):—

Courses and top-sails,.....	22 inches.
Topgallant-sails and royals,.....	22½ "
Foot of try-sails,.....	21½ "
Foot of jibs,.....	22-7 "

The tablings of the leeches of a square-sail are about one cloth in breadth; that of the foot, half a cloth; the reef-bands, one-third of a cloth.

Cloths which are cut obliquely at the ends are said to be *gored*.

The seams of square-sails run vertically; hence their cloths are not gored at the head. At the foot they are gored to the extent of the roach, if there is any; if the leeches are inclined, there are gored cloths between them and the foot.

In studding-sails, the seams usually run parallel to the inner leech. Lower studding-sails are commonly rectangular, so that there are no gored cloths in them; in upper studding-sails, the cloths are gored at the head, foot, and outer leech.

In fore-and-aft sails, the seams most commonly, though not always, run parallel to the after leech; and the cloths in most cases are gored at the head, foot, and luff. In what are called "angulated jibs," of which examples are shown in Plate $\frac{F}{4}$, there are two sets of seams, parallel respectively to the foot and to the after leech.

The foot and leech ropes of square-sails are from 0.5 to 0.6 of the girth of the shrouds of the masts to which those sails belong: the head-ropes, about half the girth of the foot and leech ropes. The bolt-ropes of fore-and-aft sails are in girth about 0.6 of the foot and leech ropes of square-sails suited for the same masts.

SECTION III.—RUNNING RIGGING.

49. The *Materials of Running Rigging* are hempen ropes and iron chains, as to the weight and strength of which, see Article 39. Wire ropes are not well suited for running rigging, because of their stiffness.

The following parts of the running rigging are almost always made of chain: *slings* of the lower yards; *topsail-tyes*, for hoisting and lowering topsail-yards; *topsail-sheets*, for hauling out the clews of the topsails. Other parts are sometimes made of chain, especially the lower and heavier parts, and those which remain nearly always taut. The chief advantage of chain is its flexibility, and consequent freedom from the waste of work which takes place in overcoming the stiffness of ropes. For the lighter and loftier parts, and for those which may have to hang slack, chain is too heavy.

50. *Blocks, Tackle, and Purchases*.—A *block* consists of an oval *shell*, usually of elm or metal, containing one or more pulleys called *sheaves*, of lignum-vitæ or metal, turning upon a wrought-iron *pin*. The round hole in the centre of a wooden sheave is lined with a gun-metal tube called the *bouching*. The part of the sheave-hole through which the rope reeves is called the *swallow*. In the bottom and sides of a block is a groove called the *score*, into which fits the *stop* or *strapping* of rope or iron by which the block is hung or secured to its place.

Ordinary blocks containing one pin are called *single*, *double*, *treble*, &c., according to the number of sheaves that turn on that pin side by side. Each sheave turns in a separate hole in the shell. The size of a block is described by its length, which is usually *three times the girth* of the rope that reeves through it. For the detailed description of various kinds of blocks, reference must be made to books on seamanship.

A combination of blocks and ropes is called a *tackle* or *purchase*. The mechanical action of a purchase depends on the principle of the equality of energy and work (stated in Article 64 of the first Division). In the absence of friction, the load lifted or useful resistance overcome by any purchase, would be to the effort exerted on the hauling part precisely in the inverse ratio of the velocities of the points of application of those forces;

but as some work is always expended wastefully in overcoming friction, the resistance overcome by a given effort is less, and the effort required to overcome a given resistance greater, than that rule would give.

The simplest purchase is a *single whip*, in which a rope reeves through a single fixed block; and the resistance (neglecting friction) is simply equal to the effort: the only use of the block being to change the direction of the rope.

In any more complex tackle, the *purchase gained* (or ratio in which the resistance overcome is greater than the effort exerted, neglecting friction) is simply equal to the *number of parts* (that is, plies) *of the rope that reeve through or lead to the running or fly block*. One of those parts is always a *standing part*: when the whole number of parts is even, the standing part is secured to the fixed block or to a fixed point near it; when odd, to the running block, or a point near it in the object to which it is fastened.

The number of sheaves in a block is half the number of parts of the rope that lead to it, if the latter number is even; and if it is odd, half the next less even number.

Sometimes the running block of one tackle has its stop secured to the hauling part of another, so as to haul upon the latter rope; and then the purchase gained is the *product* of the numbers expressing the purchase gained by the two tackles separately. In such cases, the rope which has the running block fixed to it is usually called the *pendant*, and the rope that is directly hauled upon by hand, the *fall*.

In complex combinations of tackle, the purchase gained can always be found by considering how many times faster the *hauling part* of the rope, where the effort is exerted, moves than the point where the useful work is done.

For various special names given to purchases used on board ship, reference must be made to treatises on seamanship. Amongst them are—a *single-whip*, already mentioned; a *double-whip*, for a twofold purchase; a *luff-tackle*, for a threefold purchase; a *gun-tackle*, for a fourfold purchase.

The *dead-eyes*, *hearts*, and *thimbles*, mentioned under the head of standing rigging, are blocks without sheaves; but they are not considered as purchases, because of the greatness of the friction in them.

51. *Description of the Running Rigging of a Ship*.—Plate $\frac{F}{4}$ shows most of the more important parts of the running rigging of a ship, as follows:—

- 43, 45, 47, Starboard fore, main, and mizen tacks. (Port-tacks not seen.)
- 44, 46, 48, Slings of fore, main, and cross-jack yards.
- 49, Starboard-vangs of main try-sail and spanker gaffs. (The port-vangs are not seen.)
- 50, Spanker-boom-sheets, or quarter-guys.
- 51, Peak-halliards of gaffs. (The throat-halliards are not seen.)
- 52, Peak and mizen signal-halliards. (The main and fore signal-halliards are not seen.)
- 53, Reef-points on fore and main courses, top-sails, and spanker.
- 54, Jib and stay-sail sheets (each sail has a pair, port and starboard).
- 56, Lifts (a pair to each yard).
- 57, Braces (each yard has a pair; but the starboard-braces only of the lower and top-sail yards are seen).
- 57,* Starboard upper mizen-topsail preventer-brace.
- 58, Reef-tackles of fore and main courses and of top-sails.
- 59, Clew-garnets of courses.
- 60, Clew-lines of upper square-sails.
- 61, Brails of spanker.
- 62, Tripping-line of spanker.
- 63, Spanker outhaul.
- 64, Spanker-boom topping-lifts.

To the parts of the running rigging already mentioned as not being seen in the Plate, the following have to be added, whose use will be explained further on:—Mast-head tackles, yard-arm tackles, top-tackles, mast-ropes, heel-ropes, jeers, tyes, halliards, downhauls, outhauls, inhauls, sheets, tacks, bow-lines, bunt-lines, leech-lines, slab-lines, and running rigging of studding-sails.

In running rigging, each rope is led once, twice, or any required number of times, between the yard or sail to be moved, and the point towards which it is to be moved; then led through a fixed block secured at or near the latter point, and thence led down on deck; where it is *belayed* (or temporarily secured) to *bitts*, *cleats*, or *belaying-pins*. The halliards of some of the lighter and loftier sails of ships are sometimes belayed in the tops, instead of on deck. Belaying-bitts are smaller than riding-bitts, and consist, like them, of a pair of upright posts and a cross-piece. They serve to belay the largest ropes; and are usually placed on deck, near the lower masts. Belaying-cleats are T-shaped, with a short neck, and long arms, or *horns*. They are fixed, usually with two bolts, in any position where they may be required, as inside the bulwarks, on the flat of the weather-deck, and round the lower masts, near the deck. Sometimes they are seized to shrouds, and are then Γ -shaped, and called *shroud-cleats*. A large belaying-cleat is called a *kevel*.

Belaying-pins (which are of wood, iron, or mixed metal) are moveable, and fit into holes in rails called *racks*, which are fixed in any convenient position—such as round the lower masts, inside the bulwarks, seized to the shrouds, &c. In Plate $\frac{B}{7}$, belaying-cleats are seen in various positions inside the bulwarks, and racks with belaying-pins near the chains of the lower rigging of each of the three masts. Bitts also are seen on the upper deck, near the masts.

A *toggle* is a short wooden pin, tapering towards both ends; it passes through an eye on a rope, and is used for hitching it to a larger eye in another rope, or on a sail.

52. The *running rigging of masts and jib-booms* consists chiefly of the following parts:—

The *masthead-tackles* of the lower masts, and *burton-tackles* of the topmasts, are purchases hanging from the pendants already mentioned under the head of standing rigging. Those of the lower masts are usually fourfold; those of the topmasts, threefold. They are for setting up rigging, assisting sometimes to stay the masts, and occasionally for sending heavy bodies aloft.

The *top-tackles*, for raising and lowering the topmasts, consist of the following parts:—The *top-tackle-pendants* are ropes, of which there are a pair each to the fore and main topmasts, and one to the mizentopmast. Each pendant has its standing part hitched to a bolt at one side of the lower mast cap. It then passes below a sheave in the heel of the topmast; then up again to the other side of the cap, and through a fixed single block (thus making a twofold purchase). The hauling part of the pendant goes straight down beside the mast, and ends in an eye, hooked to the running block of the *top-tackle-fall*, usually a sixfold purchase, the fixed block of which is hooked to a bolt in the main deck. The purchase gained is thus $2 \times 6 = 12$ -fold.

The *topgallantmast ropes*, for raising and lowering those masts, are usually one to each mast only. Each of those is fitted similarly to a top-tackle-pendant, except that its hauling part is led directly down to the deck, and has no additional purchase on it; so that the purchase is twofold only. The *topgallant-lizard* is

a rope with one end hitched to a hole in the royal-pole, and the other spliced round a thimble on the hauling part of the mast-rope: it keeps the mast from turning over when its head comes below the topmast-cross-trees.

The *jib heel-rope*, for hauling out the jib-boom, forms a twofold purchase: its standing part is secured to one side of the bowsprit-cap; it then passes round a sheave in the heel of the jib-boom, then back to the other side of the cap, and through a fixed block, and thence to the forecastle. The *flying-jib heel-rope* is single; it goes from the heel of the flying-jib-boom to a block at the jib-boom-end, and thence to the forecastle.

53. *Running Rigging of Square Sails and Yards*.—The lower yards are *swayed* (or hoisted) and *struck* (or lowered) by purchases called *jeers*, usually fourfold. The upper jeer-block usually hangs from the trestle-trees.

The topsail-yards are hoisted and lowered by *topsail-tyes*, usually of chain: of these the fore and main topsail-yards have usually a pair; the mizen topsail-yard, one only. The standing part of each topsail-tye is secured to the topmast-head or trestle-trees; thence it passes through a *tye-block* (single) on the topsail-yard, and up again to the topmast-head, forming a twofold purchase; then through a hanging block. To the hauling part is attached the fly-block, being the upper block of a twofold or threefold purchase leading down the channels, which latter tackle is called the *topsail-halliards*. The purchase gained is thus fourfold or sixfold. Sometimes there are a lighter and a heavier purchase, leading to the channels at opposite sides of the ship.

A topsail-tye for Cunningham's rolling yard is, as already stated in Article 37, a chain with the yard hanging in the bight of it. Both parts of the chain pass through blocks at the mast-head, and thence are led downwards, and have purchases or halliards at their lower ends, leading to the channels.

Topgallant and royal yards are hoisted and lowered by *halliards*, which are either single ropes or twofold purchases, and lead to the deck.

The *lifts* are the ropes or tackles, of which there are a pair to each yard, supporting its ends. They run from the yard-arms to the caps of the lower masts, and to blocks in the rigging of the upper masts, and are led thence downwards: the lower lifts, to the deck; the topsail-lifts, to the channels; the topgallant and royal lifts, to the tops. Fore and main lifts are usually threefold purchases; fore and main topsail lifts, twofold; crossjack, topgallant, and royal lifts, single ropes.

The *braces* are the ropes or tackles which trim the yards to various angular positions, according to the relative direction of the wind and the ship's course. Each yard has at least a pair, called respectively the starboard and port brace, or if the direction of the wind is referred to, the lee and weather brace; and the main-yard has generally a second pair, called the preventer main braces. Lower and topsail braces are double; topgallant braces generally single, sometimes double; royal braces, single. The points to which braces are usually led directly from the yard-arms are shown in Plate $\frac{F}{2}$; the hauling parts are always led down to the deck, and belayed there. The Plate does not show the *preventer main braces*, which are led to the foremast below the trestle-trees, and thence to the deck, and are alone used for trimming the yard; the *main braces*, which are led to the ship's quarters, are used for resisting the pressure of the wind only.

The clews of the courses are hauled forward by *tacks*, and aft by *sheets*, all of which are double, reeving through blocks shackled to the clews of the courses, and called *tack-blocks* and *sheet-blocks*. The fixed blocks for the fore-tacks are at the bumpkin ends, whence the hauling parts of those ropes are led inboard; the fixed blocks for the other tacks and sheets of the courses are at suitable points of the ship's sides. Sometimes a pair of *spiders*, or projecting iron arms abaft the main channels, are required in order to carry the fixed blocks for the main sheets.

The *sheets* of the upper sails are for hauling out the clews of each sail, so as to spread its foot along the yard next below. On the after side of that yard, close to the yard-arms, are bolted a pair of *cheeks*, each containing a sheave; the sheet leads from the clew of the sail through the cheek and round the sheave; thence along the after side of the yard to a *quarter-block* near the middle of the yard, and thence directly down—topsail and topgallant sheets to the deck, royal sheets to the top. Royal and topgallant sheets are usually single; topsail sheets, if of rope, double, reeving through a block shackled to the clew, and the standing part secured round the lower yard-arm: if of chain (as they almost always are), topsail-sheets are single.

Clew-garnets for the courses, and *clew-lines* for the upper sails, are ropes abaft the sails which haul their clews up to the bunt of the yards in furling. They lead from the clews of the sails to *quarter-blocks* near the middle of the yards, and thence to the deck, for all except royal clew-lines, which are commonly worked in the tops. Clew-garnets and topsail clew-lines are double, reeving through blocks shackled to the clews; the rest single.

Each square-sail (except, in most cases, the royals) has a pair of *bow-lines*, one to haul each leech to windward. The *bow-line-bridles* form a set of branching legs from the bowline to the cringles on the leech of the sail. The bowlines are led forward: those of the sails on the foremast to sheaves at convenient points on the jib-boom and bowsprit, and thence inboard; those of the upper after sails to the heads of the masts next before and below them, and thence to the deck; those of the main course to bitts on the deck.

Bunt-lines and *leech-lines* are ropes in front of a square-sail, for hauling up the foot and leeches respectively (to which they are toggled) towards the bunt of the yard. *Slab-lines* are fitted like leech-lines, but abaft the sail.

Gaskets are made of three or four strands, plaited together, and are used for tying up the furled sail.

Reef-tackles have a twofold purchase, and are used for hauling up the leeches of courses and topsails to the yard-arms before reefing the sails.

Downhauls haul the upper yards directly down; for topsail-yards they are usually twofold, and often of chain, and are essential to the working of rolling-spars.

Yard-tackles are threefold purchases, which hang from pendants at the lower yard-arms, for lifting boats and other weights. When not in use, they are triced up to the lower rigging below the futtock-shrouds.

53. *Running Rigging of Studding-sails*.—A lower studding-sail boom or swing-boom is supported at its inner end, as already stated, by a goose-neck hinged to the channels; at its outer-end, it is supported by a *topping-lift*, which leads from the boom

to the top of the lower mast, and thence to the deck or chains. The topping-lift reeves through a block or thimble on the end of the *long lizard*, a rope by means of which it can be hauled out towards the yard-arm, while the boom is being rigged out. The boom is trimmed and kept steady by ropes like braces in their position and use—the *fore-guy*, leading from the boom-end to the spritsail-gaff, thence to the heel of the bowsprit, and thence inboard; and the *after-guy*, leading from the boom-end aft to a sheave in the ship's side, and thence inboard.

A topmast studding-sail boom is rigged out and in by means of the *boom-jigger*, a twofold purchase, the running block of which is hooked to the inner end of the boom, and the fixed block to the boom-iron for hauling out, and to the top of the lower mast for hauling in.

Studding-sail halliards are usually a single rope, leading from the studding-sail yard through a block, which for lower studding-sails hangs from the topmast studding-sail boom end, and for upper studding-sails from the yard-arm above (being then called a *jewel-block*), and thence to the deck; the upper studding-sail halliards passing on the way through a block at the head of the mast next below, and the lower studding-sail halliards through a block hung by a pendant from the lower mast head. Lower studding-sails have also *inner halliards*, twofold, from the inner head cringle of the sail to the top, and thence to the deck.

Studding-sail sheets are for spreading the inner clew inwards. A lower studding-sail has two; one leading to the channels, the other inboard over the bulwarks. A topmast studding-sail has two; the *short sheet*, leading through a block on the inner boom-iron to the top; the *long sheet*, down before the course to the deck. A topgallant-studding-sail has one, led down into the top.

Studding-sail tacks are for spreading the outer clew outwards. They are single, toggled to the outer clew, rove through the *tack-block* at the boom-end, and thence led aft; the lower tacks inboard through sheaves in the ship's side; the topmast-studding-sail tacks to the deck; the topgallant-studding-sail tacks to the tops next abaft.

The *lower-studding-sail tripping-line* leads from the tack of the sail through a thimble at its centre, and a block at its inner yard-arm, to a block under the top of the mast, and thence to the deck. The *topmast-studding-sail downhaul* leads from the outer head cringle of the sail through thimbles on the outer leech, to a block at the tack, and thence to the deck. The *topgallant-studding-sail downhaul* leads from the inner head cringle of the sail to the top of the lower mast.

The fore-topmast-studding-sail booms have *boom-braces* of twofold purchase, leading to the main rigging.

54. *Running Rigging of Fore-and-aft sails, Gaffs, and Booms*. The boom of a fore-and-aft sail is hung from the trestle-trees by one or more ropes called *topping-lifts* (see Plate $\frac{F}{2}$); for a ship's spanker boom there are usually two, one at each side of the sail; and each of those two is double, being secured near the after end of the boom, thence led through a cheek on the trestle-tree, thence through a cheek near the middle of the boom, and thence forward to a threefold purchase, by which it is set up to the fore end of the boom.

The *boom-sheet* is sometimes one strong purchase, connecting the boom with the centre of the taffrail; for ship's spanker booms,

there are usually a pair of boom-sheets, sometimes called *quarter-guys*, fivefold, leading to the two quarters.

The throat of a gaff is hoisted and lowered by the *throat-halliards*, usually a threefold purchase; the lower block hooked to the throat, the upper supported by the trestle-trees; the hauling part leads down to the deck.

The peak of a gaff is hoisted and lowered by the *peak-halliards*, usually fourfold; rove through two single blocks at different points of the gaff and a double block or two single blocks at the cap or mast-head; the hauling part led to the deck.

The *vangs* lead from the gaff-end to the ship's quarters, and are usually pendants with a twofold purchase.

A standing gaff (as that of a try-sail usually is) is swayed aloft by the top-burton-tackles, and slung by two pendants; one from the throat, hooked to a bolt between the trestle-trees, and one from the peak, hooked to a bolt in the after side of the mast-head. It has vangs like a spanker-gaff.

Spankers and gaff main-sails are laced to the gaff, and bent to *hanks* or hoops on the mast, or on a try-sail mast. The tack is hauled down and forward by the *tack-tackle*, usually threefold and toggled to the tack. It can be triced up towards the throat of the gaff by the *tack tricing line*, usually a double whip. The clew is hauled out to the end of the boom by the *outhaul*, usually twofold.

The foot of a boom main-sail is sometimes laced to the boom. The *reef-tackles* haul down the reef-criingles to the boom-end, previous to reefing.

Try-sails are bent to rings upon the gaff, and the head clew or peak of the sail is hauled out to the gaff-end or in towards the throat by the *outhaul* and *inhaul*. The outhaul reeves through a sheave in the end of the gaff; then through a block at the mast-head, and is then led down to the deck with a double purchase. The inhaul reeves through a block on the throat of the gaff.

The *brails* of a gaff-sail are for hauling the after-leech of the sail forward previous to furling, towards the head (peak brails),nock (throat brails), and luff (foot brails). They are in pairs, usually five in number; each pair consists of a single rope, seized at the middle of its length to the after-leech; the two parts are led upwards and forwards, at each side of the sail, through blocks at the head, throat, or luff, as the case may be, and thence to the deck. The lee brails are hauled upon in furling.

A *trysail-sheet* for hauling aft the clew of a try-sail, is usually a threefold purchase; it is hooked to an eye-bolt at the lee side of the deck, and shifted when the ship goes about.

When there are two stays near each other, a *stay* and a *pre-venter-stay*, the stay-sail hangs from the preventer-stay. Jibs and other stay-sails are either bent to hanks which run on the stay, or are laced to the stay with a spiral lacing passing through eyelet-holes in the sail, and round the stay in the contrary direction to its lay. They are hauled up by the *halliards*, and down by the *downhaul*. The halliards are usually single or double, the downhaul single; the downhaul leads from the head of the sail, through three or four thimbles on the fore-leech, through a block at or near the tack of the sail, and thence aft. Sometimes the halliards are bent to a *head-stick*, or short yard, in length about equal to the breadth of a cloth. The jibs and stay-sails on the bowsprit and jib-booms have *tack-lashings*, to secure their tacks to the bowsprit or boom; other stay-sails

have *tack-tackles*, leading obliquely downwards and forwards, and, finally, to the deck. The clew of every jib or other stay-sail has two *sheets* to haul it aft, single or twofold purchases, leading respectively to the two sides of the vessel; except sometimes the fore-sails of cutters and smacks, which have but one sheet leading to a ring or *traveller*, that fits loosely on a round iron rod called a *horse*, stretching athwart the deck just before the mast. The tack of a cutter's jib is often secured to a ring or traveller on the bowsprit, hauled out and in by an *outhaul* and *inhaul*; and the jib has no stay. This arrangement facilitates the bending of jibs of different sizes, to suit the state of the weather.

55. *Dimensions of Running Rigging*.—The following proportions of running rigging are founded upon and condensed from the examples given by Biddlecombe and Kipping. When references are made to standing rigging, see Article 43.

Masthead-tackles: girth = girth of pendants × from 0.4 to 0.5
Jib-boom heel-rope: girth = diameter of boom × about 0.25
Mast-ropes: girth = diameter of mast × " 0.40

COURSES:—

Lifts, braces, bowlines, and bridles: girth = diameter of }
yard × " 0.25
Tacks and sheets: girth = diameter of yard × " 0.30
Clew-garnets: " = " " × " 0.22
Bunt-lines, studding-sail halliards, tacks, and sheets: girth }
= diameter of yard × " 0.20
Leech-lines: girth = diameter of yard × " 0.15
Slab-lines: " = " " × " 0.12

TOPSAILS:—

Tyes and sheets: diameter of iron for chain = diameter }
of yard × " 0.05
Halliards, braces, bunt-lines, bow-lines, reef-tackles: girth }
= diameter of yard × " 0.25
Lifts, clew-lines, studding-sail halliards, tacks, and sheets: }
girth = diameter of yard × " 0.30
Studding-sail downhauls, boom-jiggers: girth = diameter }
of yard × " 0.20

TOPGALLANT-SAILS AND ROYALS:—

Halliards and sheets: girth = diameter of yard × " 0.40
Lifts: girth = diameter of yard × " 0.35
Braces: " = " " × " 0.25
Clew-lines, bow-lines, studding-sail halliards, tacks, and }
sheets: girth = diameter of yard × " 0.22
Studding-sail downhauls: girth = diameter of yard × " 0.20

SPANKERS AND OTHER GAFF-SAILS:—

Boom topping-lifts, sheets, and guys: girth = diameter }
of boom × " 0.40
Throat and peak halliards: girth = diameter of gaff × ... " 0.40
Trysail-sheets, vang-pendants, outhauls: girth = dia- }
meter of gaff × " 0.35
Tack-tackle, tricing-line: " = " " × ... " 0.25
Brails: girth = diameter of gaff × " 0.20

JIBS:—

Halliards and sheets: girth = girth of stay × " 0.60
Downhaul: girth = girth of stay × " 0.45

STAY-SAILS:—

Halliards and sheets: girth = girth of stay × " 0.45
Downhaul: girth = girth of stay × " 0.35
(Girth of lower staysail-stay = girth of principal stay ×
about 0.4.)

56. *Positions of Tacks and Sheets*.—It is evident that, in order that tacks and sheets may bring a sail as nearly as possible into a state of uniform tension, their directions should pass through the centre of the sail, found as explained in Article 10 of this Division. This principle is usually attended to as far as practicable; and in jibs without stays, it applies to the halliards also.

NOTE TO ARTICLE 45.—The ultimate tenacity of Royal Navy canvas, as given in the Table, is equivalent to the weight of the following lengths of its own material:—Mean of Nos. 1 to 6, weft, 30,790 feet; warp, 21,550 feet: mean of Nos. 7 and 8, weft, 32,000 feet; warp, 27,200 feet.

DIVISION SIXTH.

MARINE STEAM ENGINEERING.

CHAPTER I.

OF PROPELLING INSTRUMENTS.

SECTION I.—PRINCIPLES OF THE ACTION OF PROPELLERS.

ARTICLE 1.—*General Explanations.*—Every propelling instrument, whether a paddle, a screw, or a pump, drives a ship by means of the forward reaction of the current of water which it sends backward (as has already been stated in the First Division, Articles 9 and 172). That reaction is transmitted through the propelling instrument to the ship; and when the ship moves at an uniform speed, it is equal and opposite to the resistance of the ship.

The condition of a propelling instrument is materially different from that of a surface advancing steadily through the water. In particular, the quantity of water acted upon in each unit of time, such as a second, by the advancing surface, depends on, and is proportional to, its speed through the water; whereas a propelling instrument is continually laying hold of a series of new masses of water, so that the quantity of water on which it acts in a given interval of time depends mainly on the speed of the vessel.

Of different kinds of propellers, those which have the simplest action are the *feathering paddle*, and the *jet* or *pump*; because they drive a stream of water directly astern, or nearly so. The *radial paddle* acts less simply, and less economically; because part of the stream of water on which it acts is driven obliquely upwards, and the reaction due to the vertical motion of that water has no effect in propelling the vessel. The *screw* drives water astern in various directions, more or less oblique; and the reaction due to the transverse component of the motion of each particle of water has no effect in propelling the vessel.

2. *Density of Sea-water.*—The most important constant quantity in calculations respecting the action of propellers is the *mass of a cubic foot of sea-water*; being its weight in pounds, divided by the accelerating effect of gravity in a second. When velocities are expressed in feet per second, the value of that constant is almost exactly 2. When velocities are expressed in *knots per hour*, the coefficient $5\frac{2}{3}$ is to be used instead of 2; being $= 2 \times$ the square of the number of feet per second in a knot per hour.

3. *The reaction of the stream of water* acted upon by any propelling instrument is the product of three factors: the mass of a cubic foot of water; the number of cubic feet of water acted on in a second; and the velocity, in feet per second, impressed on that water by the propeller.

The number of cubic feet of water acted on in a second, is to be calculated by multiplying the sectional area of the stream by its velocity relatively to the ship. This calculation might be made either before or immediately after the action of the propelling instrument takes place; for the increase of velocity, relatively to the ship, which the propeller produces, must be exactly balanced by a diminution in the sectional area of the stream. The former mode of calculation was that adopted in the First Division of this Treatise, Articles 172, 173, 174; but since that Division was printed, it has been ascertained that the latter mode of calculation is the more convenient; because it appears, from the results of experiment, that the sectional area of the stream, immediately after the action of the propeller, is either very nearly or exactly equal to the area of the propeller itself, projected on a thwartship plane: that is, for feathering paddles, simply the area of a pair of floats; for radial paddles, the breadth multiplied by the depth of immersion; and for a screw, the area of the disc, less that of the boss. The velocity of the stream, relatively to the ship, is the sum of two quantities—the velocity of the ship, and the velocity, relatively to still water, of the stream driven back by the propeller; in other words, the *apparent slip* of that stream.

4. For feathering paddles, the *slip of the stream* of water driven back is the same with that of the paddle-floats themselves, because they press directly backwards on the water. For a radial paddle-wheel and a screw-fan, owing to the obliquity of their action, the slip of the stream driven back is less than the slip of the paddle or screw, in the proportion of the square of the cosine of the obliquity of their surfaces to a thwartship plane: a ratio which is different for different parts of the stream of water acted on by the same paddle or screw.

The remaining factor of the reaction of the water—the velocity

impressed on it by the propelling instrument—when the propeller acts upon water that was previously still, is simply the *slip of the stream driven back*, just mentioned.

5. *Effect of working in Disturbed Water.*—If the water acted upon by the propeller has previously received any sensible velocity, either forward or backward, through the action of the ship upon it, such forward or backward velocity is to be combined with the apparent slip of the stream, in order to give what may be called its *real slip*; and that real slip is the true value of the velocity impressed on the water by the propeller. This requires special attention in the case of screws, which almost always work in water that is following the vessel.

Another circumstance also requires attention, when the propeller lays hold of water that is already in motion through the action of the vessel, viz., that the change of pressure produced in the water by the action of the propeller on it is transmitted to some part of the ship's bottom, and thus *the resistance of the ship is altered*. The alteration of resistance so produced constitutes a difference between the *total thrust* and the *effective thrust* of the propeller; and from the mathematical investigation of that sort of action (for which, see a paper by the Editor of this Treatise in the Transactions of the Institution of Naval Architects for 1865), it appears that the effect is *always to produce a waste of power*, when the propeller works in water that has been previously set in motion by the vessel; in other words, when there is a difference between real and apparent slip.

6. With regard to the *efficiency* of propellers, or the ratio borne by the useful work done in driving the vessel, to the whole work done in moving the propeller, the following results (when friction is left out of account) are applicable to all kinds of propelling instruments:—

I. When the propeller works in previously still water, there is a loss of work simply proportional to the slip of the propeller; so that the efficiency is represented by—

$$1 - \frac{\text{Slip of propeller}}{\text{Speed of propeller}}.$$

In the case of screws or other obliquely acting surfaces, the loss stated above comprehends the effects of rotatory or transverse, as well as of backward, slip of the water.

II. When the propeller works in water previously set in motion by the ship, there is, in the first place, a loss of work proportional to the *real slip* of the propeller relatively to that moving water, and then a further loss of work proportional to the *square of the previous velocity of the water*.

7. *Rules applicable to Feathering Paddles working in undisturbed water.*—RULE I.—To determine the proper area of a pair of feathering floats for the paddle-wheels of a given vessel, to be driven at a given speed, with a given slip, the paddles working in water that is not sensibly disturbed by the ship.

Calculate (according to the principles explained in the first three Sections of Chapter V. of the First Division) the probable resistance of the ship at the intended speed.

Divide that resistance by the intended speed of the centres of the paddle-floats relatively to the water (or slip), by their intended speed relatively to the vessel (or sum of the speed of the vessel and the slip), and by the mass of a cubic foot of water, viz.—for resistance in lbs., and velocities in feet per second, 2; and for resistance in lbs., and velocities in knots, 5½.

The quotient will be the required area in square feet.

RULE II.—To solve the same question, when the vessel is so proportioned that her resistance depends on her “augmented surface” only (see First Division, Chapter V., Article 162).

Divide the augmented surface by the ratio which the intended slip bears to the intended speed of the vessel, by the ratio which the intended speed of the centres of the floats bears to the intended speed of the vessel, and by the constant 566: the quotient will be the area required.

In different states of the ship's bottom the constant divisor may have different values, which, in the present state of our experimental knowledge, may be taken as ranging between 500 and 600.

EXAMPLE (from the steamer *Admiral*).

Augmented surface,.....8560 square feet.
Intended ratio of slip to speed of vessel,..... 0.28
Intended ratio of speed of centres of floats to speed of vessel,..... 1.28

$\frac{8560}{0.28 \times 1.28 \times 566} = 42$ square feet, required area of a pair of floats; giving 21 square feet for a single float. The actual floats measured 7 feet broad and 3 feet deep.

RULE III.—To find the mean moment of torsion on each paddle-shaft:—Multiply half the estimated resistance, at the greatest intended speed of the vessel, by the effective radius of the paddles, measured from the axis of the shaft to the centre of a float.

RULE IV.—To find the power required to drive the paddles:—Multiply the estimated resistance of the vessel in lbs. at her intended speed, by the speed of the paddle-floats relatively to the vessel (that is, by the speed of the vessel plus the slip), in feet per second; this will give the required power in foot-lbs. per second, *exclusive of friction*. To include friction of mechanism, from 0.25 to 0.30 of the power exclusive of friction, may be added: the sum will be the required *indicated power* of the engines.

TABLE FOR FEATHERING PADDLES.

Ratios of Slip to—		Ratio of		Efficiency of Paddles, Neglecting Friction.	Ratio of Augmented Surface of Vessel to Area of a Pair of Floats.
Speed of Ship.	Speed of Paddles.	Speed of Paddles × Slip to Square of Speed of Ship.	Speed of Paddles × Slip		
0.100	0.091	0.1100	0.909	62	Ordinary limits of practice.
0.125	0.111	0.1406	0.888	80	
0.150	0.130	0.1725	0.870	98	
0.175	0.149	0.2056	0.851	116	
0.200	0.167	0.2400	0.833	136	
0.225	0.184	0.2756	0.816	156	
0.250	0.200	0.3125	0.800	177	
0.275	0.216	0.3506	0.784	198	
0.300	0.231	0.3900	0.769	221	
0.325	0.245	0.4306	0.755	244	
0.350	0.259	0.4725	0.741	267	
0.375	0.273	0.5156	0.727	292	
0.400	0.286	0.5600	0.714	317	
0.425	0.298	0.6056	0.702	343	
0.450	0.310	0.6525	0.690	369	
0.475	0.322	0.7006	0.678	397	
0.500	0.333	0.7500	0.667	425	

In comparing this table with that given in the First Division, Article 172, page 88, it is to be observed that in the table of the First Division the area compared with the augmented surface is that of the reacting stream before being acted on by the paddles, while in the table of the present Article it is the area of the paddles themselves.*

* Provisional Rule for the Additional Resistance due to a short after-body.—As to the manner in which a short after-body acts in producing increased resistance, see the First Division, Articles 156 and 158. The following rule for estimating that resistance is to be regarded in the meanwhile as merely

The rules for the areas of feathering paddles are applicable also to the areas of jets used for propulsion.

8. *Rules for Disturbed Water.*—Although the state of motion of the water disturbed by the passage of a vessel is not yet known exactly, it can be estimated with a degree of approximation sufficient for the present purpose of computing its probable effect on the action of a propeller.

RULE I.—To calculate approximately the forward velocity of the water near the stern-post of a vessel, of which the screw lays hold: construct on the body-plan of the after-body a normal-line (see the Second Division, Article 26, page 118) passing through the centre of the screw; measure the extreme half-breadth of that line as developed on a plane surface, and divide it by the length in a fore-and-aft direction from that extreme half-breadth to the after end of the line; the quotient will be nearly the ratio of the forward velocity of the disturbed water to the speed of the vessel.

RULE II.—To find the *real slip* of the propeller: add the forward velocity of the disturbed water to the apparent slip of the propeller—that is, to the difference between its speed and that of the ship. The apparent slip of a propeller is sometimes

a provisional approximation; for it is based upon the trials of two vessels only—those designated as X and Y in Article 158 of the First Division, and elsewhere.

Compute, as in the Article just referred to, the proper least length of after-body for the intended speed; that is, take *three-eighths of the square of the speed in knots for the length in feet.*

Divide the actual length of after-body by the proper length; if the actual after-body is too short, the quotient will be a fraction less than 1; subtract the square of that fraction from 1, and extract the square root of the remainder.

Multiply that square root by the mean of the squares of the sines of the obliquities of the water-lines of the after-body to a fore-and-aft line, by the area of immersed midship section, in square feet, and by the constant 566; the product will be the *additional augmented surface*, to which the deficiency of the length of the after-body is equivalent.

EXAMPLE I.—Steamer X.

Intended speed, about 15 knots.

Proper length of after-body, $15^2 \times \frac{3}{8} = 84$ feet, nearly.

Actual length = $\frac{60}{84} = \frac{5}{7}$; and $\sqrt{1 - \frac{25}{49}} = 0.7$, nearly.

Mean of squares of sines of obliquities of water-lines of after-body,..... 0.04

Immersed midship section, 61 square feet; $0.7 \times 0.04 \times 61 \times 566 = 967$ square feet, being the additional augmented surface equivalent to the deficiency of length of the after-body.

Actual augmented surface, computed as in the First Division, Article 166, 967 "

Add for short after-body,..... 3949 "

Total,..... 3949 "

This might be called the *corrected augmented surface*.

Computation of probable speed with the actual indicated power of..... 655.5 horses.

Multiply by the ordinary coefficient of propulsion,..... 20,000

Divide by corrected augmented surface,..... 3949) 13,110,000 product.

Cube of probable speed,..... 8920

Probable speed, by calculation,..... 14.918 knots.

Actual speed, by trial,..... 15.065 "

Difference,..... 0.147 knot;

or about 1 per cent.; and the error is on the safe side.

EXAMPLE II.—Steamer Y.

Intended speed, about 17½ knots.

Proper length of after-body, $(17\frac{1}{2})^2 \times \frac{3}{8} = 115$ feet, nearly.

Actual length = $\frac{70}{115} = 0.61$, nearly; and $\sqrt{1 - (0.61)^2} = 0.79$, nearly.

Mean of squares of sines of obliquities of water-lines of after-body,..... 0.036 nearly.

Immersed midship section, 70 square feet, nearly; $0.79 \times 0.036 \times 70 \times 566 = 1127$ square feet, nearly; additional augmented surface.

Actual augmented surface,..... 3965 square feet.

Add for short after-body,..... 1127 "

Corrected augmented surface,..... 5092

Computation of probable speed, with the actual indicated power of..... 1316 horses.

Multiply by the ordinary coefficient of propulsion,..... 20,000

Divide by corrected augmented surface,..... 5092) 26,320,000 product.

Cube of probable speed,..... 5169

Probable speed, by calculation,..... 17.29 knots.

Actual speed, by trial,..... 17.43 "

Difference,..... 0.14 knot;

being about 0.8 per cent., and on the safe side.

negative; and then it must be subtracted from the forward velocity of the water in order to find the real slip.

As a preliminary step to the calculation of the area of a screw, it is useful to calculate the area of a feathering paddle, or of a jet, for doing the same work with the same slip; and such is the object of the next rule.

RULE III.—To find the area of a feathering paddle or of a jet, for driving a given ship with a given speed and a given slip. Proceed, in the first place, as in Rule I. or Rule II. of the preceding Article, using the *real slip* in the calculation. Then multiply the area so found by the speed of the ship, less the forward velocity of the water of which the propeller lays hold, and divide by the speed of the ship, less twice the same velocity; the result will be the area required.

RULE IV.—To find the *efficiency* of such a feathering paddle or jet, neglecting friction: divide the forward speed of the disturbed water by the speed of the ship, less that speed; subtract the square of the quotient from 1; multiply the remainder by the speed of the propeller, less the real slip, and divide by the speed of the propeller.

As before, the work done in a given time in driving the ship, divided by the fraction expressing the efficiency of the propeller, gives the work done in driving the propeller, neglecting friction; to which work from 25 to 30 per cent. is to be added for friction, in order to obtain the indicated work of the engines in the same time.

EXAMPLE (from H.M.S. *Warrior*).

By the application of Rule I., the probable mean speed of the water following the ship is found to be the following

fraction of the ship's speed,..... 0.09

Add apparent slip (the speed of propeller being 1.12),..... 0.12

Real slip (by Rule II.), in fractions of the ship's speed,..... 0.21

Area of propeller
Augmented surface of ship

as calculated by Rule II. of the preceding Article,
 $= \frac{1}{1.12 \times 0.21 \times 566} = \frac{1}{133}$; which has to be corrected, as follows,

by Rule III. of the present Article—

$\frac{1}{133} \times \frac{1}{1 - 0.09} = \frac{1}{133} \times \frac{91}{82} = \frac{1}{120}$, nearly.

The efficiency, by Rule IV., is calculated as follows—

$\left\{ 1 - \left(\frac{0.09}{0.91} \right)^2 \right\} \times \frac{0.91}{1.12} = 0.8$, nearly.

The Rules of this Article are modified to suit the case of a propeller working in water that is moving astern, by treating the speed of such water as negative; observing that, as the square of a negative quantity is positive, Rule IV. gives a loss of efficiency in this case as well as in that of water moving ahead.

When the paddle-wheels are situated at, or near, the extreme breadth of the after-body, the mean backward speed of the current in the water of which they lay hold may be estimated roughly, in fractions of the speed of the vessel, as follows:—

RULE V.—Divide once-and-a-quarter the extreme half-breadth by the length of the after-body: the quotient so found is to be subtracted from the apparent slip of the paddles to give the real slip, in fractions of the speed of the vessel.

RULE VI.—But if there is a straight middle, so as to place the paddles at a distance afore the extreme breadth of the after-body equal in feet to, at least, about *one-seventh* of the square of the speed in knots, the current may be treated as insensible.

There are not yet sufficient experimental data to determine how intermediate cases ought to be treated; and, indeed, the two preceding rules themselves, in the present state of our knowledge, must be regarded as provisional approximations only.

RULE VII.—To find the ratio of the area of a pair of feathering paddles to the augmented surface (corrected, if necessary, for short after-body) of a vessel, with a given ratio of apparent slip to speed, the paddles working in a backward current bearing a given ratio to the speed of the vessel—

Multiply together the real slip and the speed of the paddles, both in fractions of the speed of the vessel; multiply the product by the speed of the vessel added to twice the speed of the backward current, and divide by the speed of the vessel added to the speed of the backward current; the result, multiplied by 566, will be the divisor by which the augmented surface is to be divided in order to give the area of a pair of floats.

EXAMPLE I.—Steamer X.		Fractions of Speed of Vessel.
Apparent slip,.....		0.323
Subtract probable backward current, $\frac{9.25}{60} \times 1\frac{1}{2} =$		0.192
Probable real slip,.....		0.131
$1.323 \times 0.131 \times \frac{1.384}{1.192} \times 566 = 114$, calculated ratio; actual ratio of		
corrected augmented surface to area of floats, $\frac{3949}{35} =$	113.	

EXAMPLE II.—Steamer V.		Fractions of Speed of Vessel.
Apparent slip,.....		0.252
Subtract probable backward current, $\frac{8}{80} \times 1\frac{1}{2} =$		0.125
Probable real slip,.....		0.127
$1.252 \times 0.127 \times \frac{1.25}{1.125} \times 566 = 100$, calculated ratio; actual ratio of		
augmented surface to area of floats, $\frac{2572}{26.4} =$	98.	

The closeness of these approximations is probably in part accidental, and is not to be expected in every case.

9. *Rules for the Screw.*—The *pitch* of a screw is the length, measured along the axis, of a complete turn; and the *speed of the propeller*, in the case of a screw, means the pitch multiplied by the number of turns in an unit of time. The case of a screw having different pitches at different parts of its surface, will be considered further on. The effective area or *disc* of a screw-propeller is measured on a thwartship plane, and has for its outer boundary the circle swept by the tips of the blades, and for its inner boundary the outline of the boss. The calculation of the proper disc-area for a screw, by following theoretical principles precisely, is of great length and complexity; and for its details reference must be made to the Transactions of the Institution of Naval Architects for 1865. For ordinary practical purposes, when the circumference of the screw is not less than $1\frac{1}{2}$ times, nor more than $3\frac{1}{2}$ times, its pitch, the following approximate rule is sufficient.

RULE I.—Divide $\frac{2}{3}$ of the pitch by the circumference, and subtract the quotient from 1. The remainder expresses the ratio of the area of an *equivalent feathering paddle* to that of the screw. Then find, by the rules of Article 8, the area of a feathering paddle required to drive the given ship at the given speed with the given real and apparent slip. Divide that area by the ratio already found; the quotient will be the required effective area of the screw; and the efficiency of the screw,

neglecting friction, will be very nearly equal to that of the paddle. To allow for the friction of the screw in the water, about $2\frac{1}{2}$ or 3 per cent. may be subtracted from the efficiency, in ordinary cases.

RULE II.—To find the diameter, circumference, and pitch of a screw, whose effective area is given. From 1 subtract the square of the ratio in which the diameter of the boss is to be less than that of the screw; divide the effective area by the remainder; the quotient will be the area, including the boss. Multiply that area by 1.273 (or multiply by 14, and divide by 11); the square root of the result will be the required diameter.

That diameter, multiplied by 3.1416 (or by $\frac{22}{7}$), will give the circumference; which, divided by the predetermined ratio of the circumference to the pitch, will give the pitch.

EXAMPLE (from H.M.S. *Warrior*)

Area of equivalent feathering paddle (as already computed in the preceding Article) $1\frac{1}{3}$ of the augmented surface of the ship.

Circumference of screw = $2\frac{1}{2}$, nearly.

Pitch
 $1 - \frac{0.8}{2\frac{1}{2}} = 1 - 0.32 = 0.68$, ratio of area of equivalent feathering paddle to area of screw. Then—

$\frac{1}{120 \times 0.68} = \frac{1}{81.6}$ ratio of required effective area of screw-disc to augmented surface.

Augmented surface, 36,979 = 453 square feet, effective area.
81.6

Diameter of boss to be $\frac{1}{4}$ of that of screw; then, $453 \times \frac{16}{15} = 483$ square feet, total area.

$\sqrt{483 \times \frac{14}{11}} = 24.8$ feet, calculated diameter.

The actual diameter is..... $24\frac{1}{2}$ feet.

Calculated circumference,..... 77.91 feet; pitch, 31.16

Actual circumference,..... 76.97 feet; pitch, 30.00

RULE III.—To find the mean moment of torsion on the propeller shaft (excluding friction of bearings); multiply the thrust of the screw by the pitch, and divide by 6.2832. (The thrust may be taken, for the purposes of this calculation, at about 3 per cent. greater than the resistance of the vessel.

EXAMPLE (from H.M.S. *Warrior*).

Thrust, 86,000 lbs. \times pitch, 30 feet = 410,000 foot-lbs., nearly.
6.2832

RULE IV.—To design a pair of twin-screws, equivalent to a given single screw. Make the pitch, diameter, and all the other dimensions of the twin-screws $\frac{1}{\sqrt{2}} = 0.7071$ of the corresponding dimensions of the single screw. The twin-screws must then make $\sqrt{2} = 1.4142$ revolution for each revolution of the single screw; and the mean moment of torsion on each of the shafts will be $\frac{1}{2\sqrt{2}} = 0.3535$ of that on the shaft of the single screw.

The preceding rules are based on the supposition that the screw is free from any defects in construction or position tending to impair its efficiency. What such defects are, and how they are to be avoided, will be considered in Section II.

Experiments have been made by Mr. Rigg upon a screw fitted with an appendage astern of it, which consists of a set of radiating fixed blades, so shaped as to receive the obliquely-moving streams of water which come from the screw, and turn

those streams into a direction right aft. If the action of this apparatus were theoretically perfect, its effect would be to make the screw equivalent in thrust and efficiency to a feathering paddle of the same area.

10. *Rules for Radial Paddles.*—The sectional area of the stream driven aft by a common or radial paddle, is the product of the breadth of the paddle into the greatest depth of immersion of its lower edge.

The slip is different at different parts of that stream. For purposes of calculation, it is convenient to take the slip at the greatest depth below the surface. The apparent slip, at that point, is the excess of the speed of the outer edges of the paddles above the speed of the vessel; and it may have to be corrected, as in Article 8, Rule V., to find the real slip.

The action of radial paddles on the water is very complex. Each particle of the stream that is driven aft quits the paddle at its outer edge, and is driven obliquely upwards (with the exception of the lowest layer of particles only, which are driven horizontally). The following rules are founded on a mathematical investigation, which is confirmed, in a general way, by such comparisons as exist between the performances of the same vessels, with radial and with feathering paddles;^a but so few of those comparisons have been so conducted as to give definite results, that the rules must be regarded as provisional approximations only.

RULE I.—To find the sectional area of the stream to be driven back. Proceed as for feathering paddles, supposing the slip of the feathering paddles to be equal to that of the lower edges of the radial paddles, and having due regard to the effect which the varying immersion of the vessel may have upon the depth of the stream.

RULE II.—To find the efficiency. Multiply the efficiency of the corresponding feathering paddle by the square root of the fraction of the outer radius of the radial paddle-wheel which stands above water.

The reciprocal of that square root is, of course, the ratio in which the power required to drive the paddle wheels is greater than required to drive the corresponding feathering paddles.

For example, if a radial paddle-wheel has 0.36 of its outer radius immersed, so that 0.64 of that radius is above water, its efficiency will be nearly $\sqrt{0.64} = 0.8$ of that of the corresponding feathering paddle, and $\frac{1}{0.8} = 1.25$ times the power will be required to drive it at the same speed; the speed being measured at the centres of the feathering paddles, and at the outer edges of the radial paddles.

RULE III.—To compute the mean moment of torsion on the paddle-shaft (exclusive of friction):—Multiply half the estimated resistance of the vessel by the outside radius of the paddles, and by the square root of the ratio in which that outside radius is greater than the height of the axis above the surface of the water.

SECTION II.—CONSTRUCTION OF PROPELLERS.

11. *Strength of Shafts.*—In Articles 7, 8, 9, and 10, it has been shown how to compute the mean effective moment of torsion on the shaft, either of a screw or of a paddle-wheel.

The strength of the shaft, however, must be suited to bear, not merely the mean effective moment, but the greatest total moment of torsion; and in determining the greatest moment from the mean moment, the following rules are to be observed.

RULE I.—For a screw-propeller shaft. Add *three-tenths* to the mean effective moment for friction; this will give the *mean total moment*. Then to find the *greatest total moment*, multiply by one or other of the following factors:—

If the screw is driven by a single engine,.....	1.57
“ “ “ by a pair of engines,.....	1.11
“ “ “ by three engines,.....	1.05

RULE II.—For paddle-shafts, either in one piece for both paddle-wheels, or coupled amidships by an intermediate shaft or otherwise. In this case regard must be had to the fact, that while the ship rolls in a seaway, one paddle-wheel may occasionally be lifted wholly out of the water, so as to throw the whole power of the engines on the shaft of the other wheel. Therefore the mean twisting moment on the shaft of one paddle-wheel is, in the first place, to be *doubled*. Then add *one-fourth* for friction, and multiply by one or other of the following factors:—

If the paddle-wheels are driven by a single engine,.....	1.57
“ “ “ by a pair of engines,.....	1.11
“ “ “ by three engines,.....	1.05

RULE III.—The greatest twisting moment on an *intermediate shaft* is that due to one engine only; nevertheless the changes of stress on the intermediate shaft are so irregular and so sudden, that it is found necessary in ocean steamers to swell its diameter in the middle to about once and a sixth that of the paddle-shafts.

RULE IV.—For each shaft of a pair of independent paddle-wheels, take half the moment given by Rule II., as the case may be; observing, that the factor expressing the ratio in which the greatest exceeds the mean total moment, depends on the number of engines that drive *each independent shaft*; so that, for example, if a pair of independent paddles are each driven by a single engine, the multiplier is 1.57.

RULE V.—To deduce the *mean total moment* from the total indicated power of the engines:—Multiply the I.H.P. by 33,000 for foot-lbs. per minute, and divide by $6.2832 \times$ the number of revolutions per minute: the quotient will be the mean total moment in foot-lbs.; or otherwise, multiply the I.H.P. by 8250, and divide by the number of revolutions per minute.

RULE VI.—To find the diameter of a shaft suited to bear a given greatest twisting moment.

Reduce the moment to inch-lbs.; divide it by $0.196 \times$ the working modulus of stress in lbs. on the square inch: the cube root of the quotient will be the required diameter in inches.

For various comparisons of the diameters of shafts used in practice, with the estimated greatest twisting moments to which they are exposed, it appears that the working modulus of stress ranges from—

8,000 lbs. to 10,000 lbs. on the square inch;
so that the divisor in the rule ranges from—
 $0.196 \times 8,000 = 1568$
to $0.196 \times 10,000 = 1960$.

The higher values of the modulus are on the whole from

^a See a Paper by Mr. J. R. Napier, from data supplied chiefly by Mr. William Beardmore, in the Transactions of the Institution of Engineers in Scotland for 1863-64.

examples of paddle-shafts; the lower, from examples of screw-shafts.

Considering the comparative weakness of iron in large forgings, the preceding values of the modulus of working stress correspond to factors of safety of from $5\frac{1}{2}$ to $4\frac{1}{2}$: the smaller factors being for paddle-shafts. The durability of paddle-shafts under those circumstances, is probably due to the fact that they are only occasionally exposed to the severe stress which takes place when one wheel is lifted wholly out of the water.

EXAMPLES OF RULES I. AND VI.

Given, mean effective twisting moment on a screw shaft,	410,000	foot-lbs.
Add three-tenths for friction,	123,000	"
Mean total moment,	533,000	"
The shaft being driven by a pair of engines, multiply by	1.11	
Greatest total moment,	591,630	foot-lbs.
To reduce to inch-lbs., multiply by	12	
Taking as the working modulus 8000 lbs. on the square inch, divide by $8000 \times 0.196 =$	1568	7,099,560 inch-lbs.
Cube of required diameter,	4.528	
Required diameter,	16.58	inches.

EXAMPLE OF RULES II. AND VI.

Given, mean effective twisting moment exerted by one paddle-wheel,	23,000	foot-lbs.
Add one-fourth for friction,	5,750	"
Mean total moment for one wheel,	28,750	
	$\times 2$	
Mean total moment for both wheels,	57,500	
The paddle-wheels being driven by a pair of engines, multiply by	1.11	
Greatest twisting moment,	63,825	foot-lbs.
To reduce to inch-lbs., multiply by	12	
Taking 9000 lbs. on the square inch as the working modulus, divide by $9000 \times 0.196 =$	1764	765,900 inch-lbs.
Cube of diameter,	434	quotient.
Required diameter,	7.57	inches.

EXAMPLE OF RULES IV. AND VI.

Independent paddle-shaft for the same vessel.—		
Mean total moment for one wheel, as before,	28,750	foot-lbs.
Multiply by	1.57	
Greatest twisting moment,	451,375	foot-lbs.
To reduce to inch-lbs., multiply by	12	
Divide, as before, by	1764	541,650 inch-lbs.
Cube of diameter,	307	
Required diameter,	6.75	inches.

EXAMPLE OF RULES V. AND VI.

Indicated horse-power of engines,	5,471	
	$\times 33,000$	
Divide by	6.2832	180,543,000 { indicated power in foot-lbs. $\frac{1}{2}$ min.
Divide by number of revolutions per minute,	$54\frac{1}{2}$	28,734,000 nearly, quotient.
Mean total twisting moment,	529,600	foot-lbs., nearly.
Multiply (as in example of Rule I.) by	1.11	
Greatest total moment,	587,856	
	$\times 12$	
Divide (as in the first example) by	1568	7,054,272
Cube of diameter,	4,499	
Required diameter,	16.52	inches.

12. *Construction of Radial Paddle-wheels.*—Common or radial paddle-wheels, though less efficient than those with feathering paddles, are much used on account of their greater simplicity and less cost, and also because they are thought to be less liable to injury in rough and stormy seas.

From the principles explained in Article 10, it appears that the efficiency of radial paddles is promoted by making the radius of the wheels as large as practicable. Ordinary proportions make the radius from two and a half to three times the greatest depth of immersion of the paddles, when the vessel is fully loaded.

According to ordinary practice, the number of radial paddles is about one to each foot in diameter of the wheel; so that their average *pitch*, or distance apart on the circumference of the wheel, is about 3.14 feet; but this rule may be departed from when there is any good reason for so doing. The depth is from two-thirds to once the pitch; and this appears to be necessary in order that every particle of water may be fully acted upon. The breadth (called also the length) of the paddles depends on the area of the stream to be driven back, which is to be determined by the rules of Article 10. The paddles are usually made of elm, $2\frac{1}{2}$ inches thick, or of pine, 3 inches thick. Their backs should be chamfered or bearded to a thin edge all round, to diminish the resistance of the water to their entering and leaving it.

The framework of the wheel consists of *centres*, *arms*, and *rings*. The centres are discs or bosses—sometimes two, but oftener three, in number—keyed upon the paddle-shaft, and having sockets round their rims for the arms. The part of the shaft on which the centres are fixed is usually square, and the centres have square holes to fit it, and are fixed upon it with eight keys each. Some engineers make the shaft round, turn the holes in the centres to fit it, and fix each of them with one key; but this is considered less safe and durable than the former method.

From each of the centres there radiates a set of arms, equal in number to the paddles. Those arms are straight flat iron bars, so placed as to move edgewise through the water; their inner ends fit into holes in the centre, where they are fixed either by bolts or by cutters (wedges); the latter method appears to be the more secure. The outer ends of the arms are T-shaped, and to the cross-bars of each set of arms is rivetted an outer ring, which is of bar-iron of the same scantling with the arms. Inner rings, one or two in number to each set of arms, are rivetted to lugs, or projections from the edges of the arms.

The scantlings of the arms should be such that their combined working moment of resistance to cross-breaking may be equal to the greatest twisting moment on the shaft, exclusive of that due to engine-friction; regard being had to the fact that the rigid connection of the arms with the outer ring doubles their moment of resistance, as compared with that of similar bars fixed at one end only. If we suppose that the working modulus is the same as for the shaft, and that one-fifth of the whole moment is due to engine-friction, those principles lead to the following—

RULE.—Divide 7.5 by the total number of arms in the wheel, and by the ratio in which the depth of an arm is to be greater than the thickness: the cube root of the quotient will be the ratio which the depth of an arm should bear to the diameter of the shaft at its journals.

EXAMPLE.—Suppose the wheel to have three sets of arms, twenty-eight in each set—in all, eighty-four arms; and that the breadth of an arm is to be four times the thickness. Then—

$$\sqrt[3]{\left(\frac{7.5}{4 \times 84}\right)} = \sqrt[3]{0.0224} = 0.282;$$

being the ratio which the depth of the arms should bear to the diameter of the shaft at its journals.

The paddles are fastened to the arms immediately inside the outer ring, with bolts having a hook on one end to lay hold of the arm, and a screw and nut on the other to fix the paddle. The paddle should be guarded against being indented by the arm and by the nuts, by means of iron or steel plates on both sides of it.

In order to diminish the shocks which arise from the successive paddles striking the water, each of them is sometimes divided into two or three parts, either depthwise or breadthwise, which are placed like a series of steps. When the division is into two parts depthwise, they are fastened at opposite sides of the arms.

For the same purpose, paddles have sometimes been made with the inner corner dipping deeper into the water than the outer, so that the whole breadth does not begin to enter the water at once; and also of a spiral form.

13.—*Construction of Feathering Paddle-wheels.*—(See Plates $\frac{E}{1}$, $\frac{E}{2}$, $\frac{G}{1}$, $\frac{G}{2}$.) The proper area for the paddles of feathering paddle-wheels is determined by the rules of Article 7. The breadth (or length) is commonly from twice to three times the depth. They are usually, like radial paddles, rectangular and oblong, and sometimes have the corners rounded; and are made sometimes of wood (usually elm or pine), like radial floats, and sometimes of boiler-plate, $\frac{3}{4}$ inch or 1 inch thick. They ought always to be chamfered to a thin edge all round.

The extreme depth of immersion of the lower edge of a feathering paddle ranges from once and a quarter times to twice its depth. The *pitch* of the paddles, or distance between them, measured on the circle which passes through their centres, in some of the most successful examples, is from 1.6 times to double their depth, or nearly so; and as this appears to give a sufficient hold of the water, a closer pitch is unnecessary; although there are examples in which the pitch is but little greater than the depth.

The *mean radius* of the wheel, measured from the axis to the centres of the paddles, is seldom less than twice or more than about three and a half times the depth of the paddles. A feathering paddle-wheel continues to work efficiently at a deeper immersion, in proportion to its outside radius, than a radial paddle-wheel; having in some cases been found to answer very well with half of its outside radius immersed.

The usual mode of framing feathering paddle-wheels is exemplified in Plates $\frac{E}{1}$, $\frac{E}{2}$, $\frac{G}{1}$, and $\frac{G}{2}$. The centres and sets of arms are in general two in number only, and the arms in each set are equal in number to the paddles; so that the total number of arms in a wheel is usually double the number of paddles. The arms are commonly made to stand obliquely to the axis (as shown in the cross-section, Plates $\frac{E}{2}$, $\frac{G}{2}$), so that they lie in two “dished” or concave conical surfaces; this enables the bearing for the paddle-shaft journal, which is carried by a bracket, or projecting part of the vessel’s side, to be brought nearer to

the centre of gravity of the wheel than would otherwise be possible.

The rule for the scantlings of the arms is the same with that given in Article 12.

There is sometimes only one ring to each set of arms, immediately inside the circle of paddles, as shown in Plate $\frac{E}{2}$; and then the outer ends of the arms form curved *spurs*, to carry the bearings of the paddle-journals. In other examples (as in Plate $\frac{G}{1}$) there are two rings to each set of arms, one inside the circle of paddles and one outside, the intermediate parts of the arms being straight, and having short spurs projecting from them. The latter construction is useful in rough and stormy ocean navigation; because in the event of the feathering paddles being destroyed or disabled, radial paddles can be fixed on the parts of the arms between the two rings.

The arms are diagonally braced, as shown in the cross-section, Plate $\frac{E}{2}$.

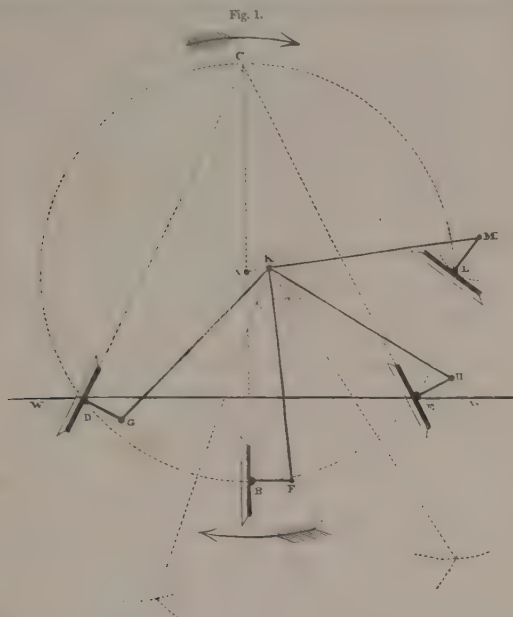
Each of the paddles has fixed to its back two steel journals, turning in holes in the spurs already mentioned, which holes are bushed with lignum-vitæ, or other wood of similar quality. From the back of each paddle projects a lever called the *stem*, or *stem-lever*, of a length equal to about *six-tenths* of the depth of a paddle. The ends of the stem-levers are connected, by means of *guide-rods* of equal length, with the *eccentric collar*; which turns freely, either about a centre-pin fixed to the spring-beam at the outer side of the paddle-box, or about an eccentric disc fixed to the side of the ship, and large enough to contain within it the paddle-shaft and its bearings. All the guide-rods, except one, are jointed by means of round pins to the eccentric collar from which they radiate: that one, called the *driving-rod*, is rigidly fastened to the collar by bolts, or otherwise, and serves to make the collar revolve once for each turn of the paddle-wheel. This is the mechanism used in “Galloway’s” (otherwise called “Morgan’s”) feathering paddle-wheel, and is the only kind of feathering mechanism that is extensively used in practice.

The following is the best construction for finding the centre about which the collar with the guide-rods ought to turn. (See Fig. 1.)

Let A represent the centre of the paddle-wheel; BC, the vertical diameter of the circle in which the centres of the paddle-journals revolve. The most efficient kind of motion for the paddles to have is such, that in going into and coming out of the water, they shall move edgewise *relatively to the current which their action on the water produces*; that is, relatively to a current moving horizontally astern with the velocity of the point, B. In order that such may be the motion of the paddles, their surfaces during their action on the water should be parallel to lines radiating from C. Therefore, draw WW to represent the load-water-line, cutting the circle, BC, in D and E; then draw the straight lines, CD, CE; these will show the proper positions for paddles leaving and entering the water respectively. The line CB already shows the proper position for a paddle at its deepest immersion, which is vertical. Draw DG = BF = EH = about 0.6 of the depth of the paddle, to represent the three corresponding positions of the stem-lever, which is commonly perpendicular to the paddle, but may, if convenient, be placed slightly oblique to it. Then find the centre, K, of a circle traversing the three points, G, F, H; that point will be the proper position for the centre of the collar; and KG =

$KF = KH$, will each be equal to the radius of the circle of pins in the collar added to the length of a guide-rod.

To make sure whether the guide-rods will at all times clear the inner edges of the floats, produce AK to M , making $KM = KG$; about M , with a radius equal to the length of a stem-lever, draw a short circular arc cutting the circle, BC , in L ; then draw a paddle in the position corresponding to the position, LM , of the stem-lever; if the guide-rod, KM , clears the



paddle in that position, it will clear it in every other. Should it be necessary to make the paddles so deep, or the stem-levers so short, that the edge of the paddle encroaches on the guide-rod, KM , the floats may be notched at the proper place, so as to clear the rods.

When the feathering machinery is constructed according to those rules, the paddles, although they do not perform the required motion with mathematical precision, approximate to it nearly enough for practical purposes throughout the arc, EBD , and for a short distance beyond it at both ends.

When the paddles are carried by spurs which project, like those shown in Plate $\frac{5}{2}$, each spur, at its junction with the ring, should have its depth regulated by the following rule—

RULE.—Divide the number of paddles in a wheel by the ratio in which half the mean radius of the wheel is greater than the projection of the spur beyond the ring in a radial direction; the square root of the quotient will be the ratio in which the spurs should be deeper than the arms, supposing them of the same thickness. For example—suppose the number of paddles in a wheel = 10; half mean radius of wheel = $3 \times$ projection of spur; then $\sqrt{\frac{10}{3}} = 1.826$, is the ratio in which the root of the spur should be deeper than the arm.

The stem-levers may be made about half the depth which the preceding rule gives for the spurs. The greatest stress upon a stem-lever and its guide-rod occurs when the entering and leaving floats are about half immersed, as at D and E , Fig. 1; and then, according to the ordinary way of fitting up

the mechanism, there is thrust on the guide-rods when the vessel is driven ahead, and tension when she is driven astern. Mr. Scott Russell fixes the stem-levers on the fronts instead of the backs of the floats, and places the centre of the eccentric collar abaft instead of afore that of the paddle-wheel, so that there is tension on the guide-rods when the ship is driven ahead, and thrust when she is driven astern; being an arrangement more favourable to the strength and durability of the rods than the ordinary arrangement.

Every paddle-wheel, feathering or radial, should be accurately balanced on its axis. It is a bad practice to fix counterpoises in the paddle-wheels, for the purpose of balancing the reactions of the machinery inboard; for such counterpoises produce great and irregular strains, and have been known to cause the machinery to break down. The machinery inboard should have its reactions independently balanced, as will be explained in a later chapter.

14. *Paddle-boxes* are so proportioned that the paddles have a clearance edgewise and endwise of about $\frac{1}{8}$ of their depth.

The outer side of a paddle-box is supported by the *spring-beam*, which lies parallel to the ship's side, and has its two ends carried by the projecting ends of a pair of thwartship beams called the *paddle-beams*, which lie parallel to each other completely athwart the vessel, one afore and the other abaft the paddle-wheels. For examples of paddle-beams, see the longitudinal section of the *Persia*, Plate $\frac{A}{2}$, also Plate $\frac{B}{2}$.

The strength required in paddle-beams and spring-beams depends on the way in which the paddle-wheels are supported.

The shaft of a radial paddle-wheel rests upon two bearings; one is near its inboard end, supported on the entablature or uppermost part of the engine-framing; the other, according to one mode of construction, is at the outboard end of the shaft, and is supported by the spring-beam. The length of each of those bearings is from once to once and a quarter the diameter of the shaft.

According to that construction, the principal part of the load upon the projecting end of each paddle-beam is one-half of the load on the outboard bearing of the paddle-shaft; which load is usually equal to, or a little less than, the weight of the paddle-wheel. When a detailed design has been made for the paddle-wheel, that weight can easily be calculated with precision; but before such a design is made, it may be approximately estimated as being for radial paddles about *five times the weight of the outboard part of the paddle-shaft*, treated as cylindrical. Hence follows—

RULE I.—To calculate approximately that part of the bending moment on a paddle-beam which arises from the weight of the paddle-wheel and shaft: multiply the square of the diameter of the shaft in inches by the length of its outboard part in inches, and divide the product by 4000; the quotient will be the load in tons, which, being multiplied by the outboard projection of the paddle-beam in inches, will give the required bending moment in inch-tons.

EXAMPLE.

Diameter of shaft, 22 inches; $22^2 =$	484
\times length of shaft outboard,	135 inches.
Divide by	4000) 65,340 product.
Approximate load on end of paddle-beam,	16.335 tons.
\times leverage,	135 inches.
Approximate bending moment,	2205 inch-tons.

The dimensions of wooden paddle-beams, when compared with the moments of their loads calculated as above, usually correspond to a working modulus of stress of about *half a ton on the square inch*. Hence follows—

RULE II.—To find the depth of a square wooden paddle-beam: multiply the bending moment in inch-tons by 12; the cube root of the quotient will be the required depth in inches.

EXAMPLE.

Moment, as before,.....	2205 inch-tons.
	× 12
	26,460
Cube root = required depth,.....	29.8 inches.

Large iron paddle-beams are usually box-girders, like those whose cross-sections are shown in Plates $\frac{A}{2}$ and $\frac{B}{2}$. The working modulus of stress in actual examples is found to range from $2\frac{1}{2}$ to 5 tons on the square inch; the former value being probably too small, and the latter rather too large. It is probable that 4 tons on the square inch is a good value for practice. The depth of the inboard part of the beam, which ought to be uniform, is usually about *one-fifth* of the outboard leverage of its load, and is sometimes diminished towards the outboard ends; and the material in most cases is so distributed, that the top, bottom, and sides contain each about one-fourth of the whole sectional area; in which case the moment of resistance is the same as if half the material were concentrated at one-third of the whole depth above the neutral axis, and half at one-third of the whole depth below. Hence follows—

RULE III.—To find the effective sectional area of iron required in a paddle-beam of a given depth: divide the bending moment in inch-tons by $\frac{2}{3}$ of the depth in inches.

EXAMPLE.

Bending moment, 2205 inch-tons; depth, 27 inches; $\frac{2}{3} \times 27 = 36$; then
$\frac{2205}{36} = 61.25$ square inches, required effective sectional area of iron;
of which one-fourth is to be put into the top, one-fourth into the bottom, and one-fourth into each side.

RULE IV.—To find the bending moment at the middle of the spring-beam: multiply the load at the end of a paddle-beam by half the span from centre to centre of the paddle-beams.

EXAMPLE

Load, as before,.....	16½ tons.
Half span, 21 feet 8 inches =	260 inches.
Bending moment required,.....	2247 inch-tons.

When the spring-beam of a large steamer is of wood, it must in general be built or trussed in order to give the requisite strength; and for that the outer side of the paddle-box affords ample depth. When of iron, it may be a box-girder or an I-shaped girder, of the depth required for strength (for example, double the depth of the paddle-beam) in the middle, and gradually diminishing, with a curved outline, to the depth of the paddle-beam at its ends.

The breadth of the spring-beam ought to be not less than two-thirds of the depth of the paddle-beam, in order that it may have strength and stiffness enough to bear lateral shocks.

When the shaft of a paddle-wheel is supported by a bracket, as in Plate $\frac{E}{2}$, the paddle-beams and spring-beam have to carry only the feathering gear (if any) and the outer part of the paddle-box and wings, and are made about half the dimensions every way of those for radial paddle-wheels of the same size.

The wings, or nearly triangular platforms which project from

the ship's sides afore and abaft the paddle-boxes, are supported by brackets, or projecting beams, usually spaced from $2\frac{1}{2}$ to 3 feet from centre to centre. The depth of each of those brackets, at the ship's side, is usually about one-fifth of its overhanging length, diminishing to two-thirds or one-half of that depth at the outer end. When of iron, the bracket commonly consists of a plate web of $\frac{3}{8}$ -inch thick, or thereabouts, with angle-iron flanges along the edges and ends, each of a sectional area equal to about half the greatest sectional area of the web: the flange at the inner end is riveted to the ship's side. The fenders, forming the continuation of the spring-beam along the outer edges of the wings, are usually of a depth equal to that of the outer ends of the brackets added to the thickness of the planking or flat of the wings (viz., from 3 to 4 inches). They are scarfed to the spring-beam at their midship ends, with an up and down scarf, and bolted to the ship's side at their foremost and aftermost ends.

In wooden ships, brackets for supporting the wings are sometimes formed by projecting ends of deck beams, connected with the side by iron knees; and the outer ends of these are sometimes supported below by oblique struts spreading from the ship's side, called *sponsons*, and covered with a skin of plank. Sometimes the paddle-beams and fenders form the sole support of the wings.

The wings are sometimes partially and sometimes wholly covered with a deck, guarded by bulwarks; and having deck-houses upon it for various purposes. Sometimes they are covered with a grating, consisting of planks 3 or 4 inches thick, and 4 or 5 inches broad, with spaces about 2 inches wide between, to let the sea through.

The paddle-beams and wings are often connected with the ship's sides by diagonal iron stays, whose diameter is about $\frac{1}{10}$ of their length. Their use is to hold down the wings and paddle-box framing against the blow of a sea striking them from below.

15. *Parts, Figure, and Dimensions of Screw-propellers.*—As to the pitch, circumference, and disc-area, see Article 9. The following statements have to be made in addition:—

I. The forms of screw-propeller in general use consist of a boss fitted on the end of the propeller-shaft, with two or more blades arranged symmetrically round it, so as to balance each other. Single-bladed screws have been proposed, but are bad; because the weight and inertia of the single blade being unbalanced, tend to produce vibrations, and to overstrain the propeller-shaft and the framing of the ship. The shaft sometimes passes through the boss, and has an *after-bearing* in the after-stern-post, or rudder-post; when there is no such bearing, the screw is said to *overhang*: sometimes an overhung screw projects abaft the rudder, the shaft passing through an oval eye in the rudder-stock. The boss, if of small diameter (say about twice the diameter of the shaft) is usually cylindrical; if of large diameter (say from one-fourth to one-half of the diameter of the screw), spherical or spheroidal. Should a large boss be made cylindrical, it is advisable to give it spherical ends.

II. It has been proved that a large spherical boss (as in "Griffith's Screw," Plate $\frac{A}{2}$) of from one-fourth to one-third of the diameter of the screw, and sometimes even more, is favourable to efficiency; for it meets with very little resistance from the water; and it occupies the place of a part of the screw-blades which meets with much resistance and has little effect in

propulsion. It is also practically convenient, because its blades can be made separate from it, and are then easily fixed and unfixed, and replaced when lost, and may also have their position adjusted, so as to alter the pitch slightly, if required.

The centre of the boss has usually a round hole through it, turned to fit the shaft, upon which it is fastened with keys, wedged endwise into longitudinal grooves; and sometimes with a transverse bolt in addition to the keys.

III. The most common number of blades in the screws of ships of war is two; of merchant ships, three or four. Every time a screw-blade passes the stern-post, there is a slight shock given to the after part of the ship, tending to make it vibrate horizontally. This is caused by the impediment offered by the stern-post to the water which the screw carries along with it. When the screw has an even number of blades, as two or four, each pair of opposite blades produces a pair of shocks in opposite directions at the same time, which partially counteract each other's effects; when the number of blades is odd, as three, shocks take place in opposite directions alternately, and the vibration produced is greater than with an even number of blades.

IV. A screw of two blades can be disconnected from the shaft, and drawn up into a *screw-well* in the stern of the vessel, when she is to go under canvas alone. When the screw is simply disconnected from the engine, and allowed to revolve freely in the water, Mr. Scott Russell by experiment found that the loss of speed in a vessel under sail was $\frac{1}{3}$ knot out of 10 knots; showing that between 14 and 15 per cent. of the whole resistance was produced by the dragging of the screw and the friction of its bearings. Another mode of lifting the screw out of the water (adopted by Mr. Russell) is to have an universal joint in the shaft, allowing the screw to be hauled up into the stern; the moveable part of the shaft works in a narrow longitudinal well amidships.

V. The length of a single blade of the screw is measured parallel to the axis of the screw; and its proportion to the pitch is also the proportion borne by the arc of the disc occupied by that blade to a whole circumference.

The aggregate length of all the blades of a screw is the sum of their several lengths measured parallel to the axis; and its proportion to the pitch is also the proportion borne by the whole angular space on the disc occupied by the blades to a whole circumference.

It is known by practical experience, (and also by the experiments of MM. Bourgois and Moll, for an account of which in English see Mr. Bourne's Treatise on the Screw Propeller), that when the number of blades does not exceed four, the most advantageous aggregate length lies between 0.45 and 0.27 of the pitch; and that in ordinary cases its value is about one-third of the pitch; also that such aggregate length may be divided equally amongst the blades, whether two, three, or four in number.

When the number of blades is increased beyond four, the same aggregate length is no longer sufficient.

VI. In the common screw, the length at every part of each blade is the same; so that the blades, when projected on a fore-and-aft plane, appear of a rectangular figure; except that the corners are in general rounded, that having been found to save power.

In the form of blade arrived at by Mr. Griffith by means of numerous and varied trials, the length is not uniform at different points of the same blade, but is diminished towards the tip and towards the root. For example, in the two-bladed screw shown in Plate $\frac{9}{7}$, the lengths are as follows:—

At the tip, about,.....	0.07 of the pitch.
At the longest part, being about $\frac{1}{10}$ of the radius from the axis,.....	0.167 “
At the root, about,.....	0.11 “
Mean length of one blade,.....	0.12 “
Mean aggregate length of the two blades,.....	0.24 “

the outlines being rounded, as the plate shows. The effect of this on the present screw, whose circumference is $2\frac{1}{2}$ times its pitch, is to make the *breadth* of the blade, as projected on a thwartship plane, nearly the same at the tip and at $\frac{1}{10}$ of the radius from the axis, and equal to *one-sixth of the pitch*, nearly.

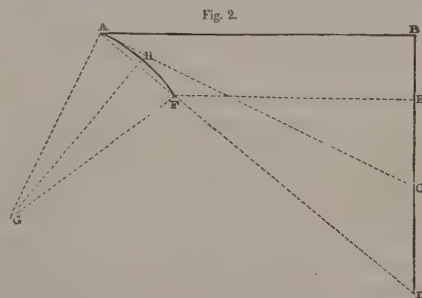
The tips of the blades are slightly bent towards the vessel, so as to present a somewhat convex surface to the water they act upon. This is better seen in one of the spare screw blades which is shown in Plate $\frac{9}{7}$, standing upright in a socket on the lower deck, a little way ahead of the fore funnel.

VII. In order to work with the greatest possible efficiency, the screw should have ample clearance in its aperture both afore and abaft. For example, in Plate $\frac{9}{7}$, the length of the screw aperture is *double the greatest length* of the screw. The chief advantage of increasing the number and diminishing the length of the blades of a screw appears to be to obtain more clearance. The stern-post and rudder-post, or, if a balanced rudder is used, its forward edge, should be bearded, to let the water move freely towards and from the screw. The top of a screw should as far as possible be well immersed.

VIII. Mr. Woodcroft's gaining-pitch screw, in which the pitch gradually increases from the leading to the following edge, has greater efficiency than the screw of uniform pitch; but in what proportion it is difficult to state with precision.

The following is theoretically the best way of designing a gaining-pitch screw.

In Fig. 2 suppose the paper to represent the development upon a flat surface of a cylinder of the circumference A B,



described about the axis of the screw. Let B C (B being forward and C aft) be the distance run by the ship while the screw makes one revolution (being the pitch of an imaginary screw turning with the real screw, but working in a solid). Join C A; then, if A is the leading edge of a blade, A C should be a tangent to that edge, which ought to touch the imaginary screw of the pitch B C, in order that it may gently cleave the water without striking it. Produce B C, and in it

take any distance, BD , that may be determined upon as the *mean* pitch of the screw-blades. Let BE be the length of the blade parallel to the axis; draw EF parallel to BA , cutting AD in F ; then F is the following edge of the blade; that is to say, the leading edge and following edge are to lie in one screw-surface of the uniform pitch BD . Draw AG perpendicular to AC ; bisect AF , and through the point of bisection draw HG perpendicular to AD , cutting AG in G . About the centre G draw the circular arc AHF , which will touch AC ; that arc will be the proper section for the screw blade upon the cylindrical surface whose circumference is AB ; and in the same manner a series of sections may be found at a series of cylindrical surfaces.

Supposing that a screw thus designed were executed with absolute precision, and that all the particles of water moved by it were acted on similarly to those in contact with the blades, *its efficiency would be equal to, and its thrust double of, that of a screw of the same area and of the uniform pitch BD , turning at the same speed, and with the slip CD at each turn.* But it is very doubtful whether so good a result as this can be practically realized.

IX. Rounding off the corners of screw-blades, and making them so that their edges present continuous convex curves, whether projected on a fore-and-aft plane or on a thwartship plane, diminishes the risk of their getting fouled by floating bodies.

X. In order to diminish the resistance to the motion of a screw-blade through the water, its edges should be thin, and its surfaces smooth. As the figure of the face of the blade, or surface which presses on the water in driving ahead, is determined by the pitch and other dimensions, the thin edges are to be obtained by giving a suitable convex form to the back of the blade, so that the thickness may gradually diminish from the middle to the edges. The thickness in the middle depends on conditions of strength, which will be considered in the next Article. To maintain smoothness of surface, bronze and yellow-metal screws should be polished, and iron screws should be coated with zinc and burnished.

XI. In vessels with a pair of *twin-screws*, one screw should be right-handed and the other left-handed; so that being turned in contrary directions in driving ahead, they may counteract each other's tendencies to produce lateral vibration.

For the description of endless varieties of forms of screw-propeller, and experiments as to their action, reference must be made to Mr. John Bourne's work already cited.

16. *Strength of Screw-propellers.*—The conditions to be fulfilled by a screw-blade as to strength are the following:—First, it should be strong enough, at least, to bear its share of the twisting moment on the shaft; and Secondly, it should be *weaker* than the shaft against the straining action of a blow of a hard body upon the tip of the blade; and the moment of resistance of the blade, in going from the boss towards the tip, should diminish faster than the moment of such a blow; that is, faster than the distance from the tip, to which that moment is proportional. This latter condition (the importance of which has been pointed out and acted upon by Mr. Scott Russell and Mr. J. R. Napier) is laid down in order that if the tip of the blade strikes against a hard substance, the fracture, if any, may take place neither in the shaft, nor in the blade near the boss, but as near as possible

to the tip; so that the broken blade may still be useful for propulsion, and capable of being repaired.

Those conditions may be fulfilled in the following manner. Suppose the developed cross-section of the blade, as made by a cylindrical surface, to be of the general character of that represented in Fig. 3, AB being the face, and ACB the back; the curve of the back being nearly a semi-ellipse, and the transverse strength, or moment of resistance, nearly the same with that of an elliptical cross-section of the same thickness. Then considering—First, that the moment of resistance of a circular or elliptical section to bending is half of its moment of resistance to twisting, and Secondly, that the working modulus of stress for the material of the blades is probably about half of that for the material of the shaft (say from 4000 to 4500 lbs. on the square inch for the blades, and from 8000 to 9000 lbs. on the square inch for the shaft), the following rule will give the proper relation between the scantlings of the shaft and blades.

RULE I.—Multiply the cube of the diameter of the shaft by 4, and divide by the number of blades, and by the length of a blade at its root, measured parallel to the axis; the square root of the quotient will be the greatest thickness of a blade *at the axis of the shaft*, supposing the blade to be continued inwards so far.

The aggregate strength of the blades against the reaction of the water will then be equal, or nearly equal, to that of the shaft, and the strength of any one blade will be a fraction of that of the shaft.

When the blades are made of the same material with the shaft, the multiplier is two instead of four.

If the cross-section of the back of the blade is to be parabolic, as in Fig. 3A, instead of semi-elliptic, as in Fig. 3, add *one-eighth* to the thickness as given in the rule for the semi-elliptic section.

Then to find the greatest thickness of the blade at other points, proceed as follows—

RULE II.—In Fig. 4 let the point A be the axis of the shaft, the circle $G D H$ the outline of the boss on a thwartship plane, and AB a radius drawn from the axis to the tip, B , of a blade. Perpendicular to AB draw AC , equal to the thickness found by Rule I., and draw the straight line $C D B$. Then the perpendicular distance of DB from GB at any point will represent the proper thickness for the middle of the blade at the corresponding point, measured in a direction normal to its face; and if this rule be observed, the moment of resistance of the blade will diminish more rapidly in going from G towards B , than the moment of a force applied at B .

Should it be considered not advisable to reduce the thickness of the blade at the tip below a certain least thickness, set off the least thickness, BE , at the tip, and through E draw a straight line parallel to $B G$; then connect the two straight lines from E and D with each other at F by means of a short curve. The point F will be that at which the blade will pro-

Fig. 3. Fig. 3 A.

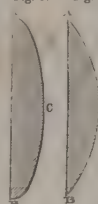


Fig. 4.



bably break, on receiving a blow from a hard body at or near the tip.

EXAMPLE.

Suppose diameter of shaft =	16 inches.
Length of root of blade parallel to axis =	40 "
Number of blades,	2

Then by Rule I.—

$$\frac{16^3 \times 4}{2 \times 40} = 204.8; \text{ and } \sqrt{204.8} = 14.3 \text{ inches, nearly} = AC \text{ in Fig. 4.}$$

Suppose further that radius of boss, $AG = \frac{1}{2} AB$; then, by Rule II., the thickness of the middle of the blade at G is found to be 10.7 inches, diminishing uniformly towards the tip.

17. *Fittings and Supports of Screw-propeller Shaft.*—The shaft of the screw-propeller passes inboard through the *shaft-pipe*, or *shaft-tube*, which in iron vessels is of cast-iron, and in wooden vessels of galvanized cast-iron or of yellow metal. The shaft-pipe is accurately turned and bored inside at and near the ends, that the shaft may rotate in it easily. In wooden vessels it extends through the stern-post and deadwood, in which a hole is bored to receive it. In iron vessels it extends through an eye in the stern-post, and through the part corresponding to the deadwood; the frames have circular arcs formed in them to curve round the pipe. The forward end of the pipe passes through a water-tight bulkhead, and contains a stuffing-box.

From that bulkhead to the water-tight after-bulkhead of the engine-room, extends a compartment containing the shaft, called the *shaft-alley* or *tunnel*; it is wide enough and high enough to allow workmen easy access to the shaft; for example, in Plates $\frac{B}{6}$ and $\frac{B}{7}$, where the shaft and ship are both large, the shaft-alley is from 6 to $8\frac{1}{2}$ feet high, and 9 feet broad; but smaller dimensions are sufficient in smaller vessels. The sides of the shaft-alley are sometimes water-tight bulkheads, and its roof is a water-tight platform. Frequently the bottom of the tunnel also is a water-tight platform.

In some vessels, the thrust of the propeller-shaft in driving ahead is communicated to the ship by means of a flat steel pivot at the forward end of the shaft pressing against a step composed of four or five bronze discs, shaped alternately like convex and concave lenses; the box containing the discs being pressed aft against the pivot by means of an adjusting wedge. In driving astern, the thrust of the screw is exerted through a pivot at the after-end of the shaft, against an elm or lignum-vitæ step in the rudder-post.

According to the construction which is now the most common, the thrust of the shaft, whether exerted ahead or astern, is communicated to the ship by means of a journal of a length equal to about twice the diameter of the shaft, and having a number of fillets or collars projecting from it, and fitting into corresponding circular grooves in a bronze bush that embraces the shaft-journal. That bush is carried by a strong plumber-block, securely framed to the bottom of the ship. The plumber-block is so placed as to be easily accessible for the purpose of lubricating the bearing. The collars are usually about an inch broad and an inch deep, and have an inch clear space between them.

Another way of receiving the thrust of the screw is to let its boss bear against a collar which surrounds the eye of the stern-post, and which has a bearing surface of wood with the fibres endwise to the thrust, formed by driving a circle of elm

or lignum-vitæ wedges into a ring-shaped groove. The water lubricates this bearing sufficiently.

When the propeller-shaft, as is often the case, is in several lengths, they are usually coupled together by means of flanges and bolts, as exemplified in Plates $\frac{B}{6}$ and $\frac{B}{7}$. To find the *proper diameter for the bolts*: divide the cube of the diameter of the shaft by the radius of the circle in which the centres of the bolts lie, by the number of bolts, and by 3; the square root of the quotient will be the required diameter of each bolt-spindle, measured inside the thread. For example, let the diameter of the shaft be 16 inches; the number of bolts 8, and the radius of the circle in which their centres lie, 12 inches; then—

$$\frac{16^3}{12 \times 8 \times 3} = 14.22; \text{ and } \sqrt{14.22} = 3.77 \text{ inches,}$$

required diameter of bolt-spindles. The weight of such shafts is supported by additional bearings, at the rate of one to each division of the shaft.

The shaft-pipes for a pair of twin-screws are sometimes made to pass through the ship's skin at suitable points in the after-part of her bottom, and supported at their after-ends by iron brackets; and sometimes they are contained in a pair of twin stern-posts and twin runs, the ship being so designed as to have a single fore-body and a double after-body.

When the after-end of a shaft-pipe is supported by a bracket, the bracket may consist of an upright arm, hanging downwards from the ship's quarter, and a thwartship arm; the arms should be in the plane either of the stern-post or of a thwartship bulkhead. They should be broad in a fore-and-aft direction, and thin transversely; if sharp-edged fore-and-aft, they will occasion the less resistance. It is difficult to lay down any precise rule for their scantlings; but a sectional area equal to about half that of a screw-blade at its root, ought to be sufficient for each of them.

(On the subject of twin-screws, see a paper by Captain Symonds, R.N., in the Transactions of the Institution of Naval Architects for 1864.)

18. *Bearings of Paddle-shafts and Propeller-shafts.*—The bearing parts of shafts, usually called *journals*, are supported by *plumber-blocks* of cast-iron, lined on the inside with *bushes*, which are renewed from time to time as they wear out. The material of a bush should be hard enough and tough enough to last well, and yet not so hard or tough as the material of the journal that turns in it, in order that the bush may wear out, and not the journal. The bush should also be of such a nature as to attract the lubricating material; hence metal bushes are suitable for lubrication with oil, and wooden bushes for lubrication with water. To make a wooden bush, pieces of lignum-vitæ or of elm are wedged into a suitable socket, with their fibres placed endwise towards the pressure; that is, the rubbing surface consists of ends of fibres.

As to the bronze used in bushes, see Division Fourth, Article 11, page 177. Cast-iron of a tough quality is suitable for the same purpose, but should not be used under water.

Soft metal for bearings (sometimes called "Babbitt's metal") has already been referred to, as a sort of metallic grease, in Article 12 of Division Fourth, page 177. It is much employed in bearings of screw-propeller shafts, and is used in the following way: a bronze or cast-iron bush is prepared, with a recess about $\frac{1}{4}$ inch deep in its bearing surface, bounded at the ends by

ledges to prevent the soft metal from escaping; the soft metal in a melted state is run into that recess, either round a core of the shape and size of the journal, or round the journal itself. Mr. Bourne recommends as a useful precaution, that the metal bush should have at its ends enough of bearing surface to bear the journal in the event of the soft metal being melted out.

The extent of bearing surface of a journal ought to be such, that the intensity of the pressure on it shall never exceed a limit, which is the lower, the higher the speed of the rubbing motion. The rules for fixing that limit which are followed in ordinary mechanism will be stated in the next Chapter; but they are not followed in the case of paddle and propeller shafts, whose area of bearing surface is made exceptionally great, for the sake of steadiness of motion and safety against heating.

The diameter of every shaft-journal is fixed with a view to strength, as explained in Article 11. Sufficient bearing surface is obtained by properly regulating the *length* of the journal.

The journals of paddle-shafts (as already stated) are usually of lengths equal to from *once to once and a quarter* of their diameters. The journals of screw-shafts, on account of their comparatively great speed of turning, are usually made of lengths equal to *double* their diameters, or thereabouts; and this rule applies especially to the main journal (mentioned in the preceding Article), with collars through which the thrust is transmitted.

It is common to insert a long bush or tube, coated on the inside with soft metal, into each end of the shaft-pipe. To keep those

bearings cool, water is allowed to leak slowly through the shaft-pipe: it should not be suffered to collect in the shaft-alley, but pumped out as fast as it comes in. The main bearing with its collars is kept cool by means of a constant circulation of water through two annular chambers encircling the shaft at the ends of the plumber-block, kept up by means of a small pump through a supply-pipe and a discharge-pipe.

All bearings, except those immersed in the water, require a constant and regular supply of oil, which is often conveyed to them through small pipes, by means of a forcing pump.

To prevent end-play in paddle-shafts when the vessel rolls, their journals are often made with one or two fillets or collars, fitting into corresponding grooves in the bushes, or bearing against shoulders. Mr. Bourne recommends instead, that those journals should be made spheroidal—that is, slightly barrel-shaped or swelled in the middle; and it is probable that a better form still is that of a spherical zone or belt, which not only prevents end-play, but accommodates itself to the drooping that almost always takes place in the outboard-end of each paddle-shaft.

Every bush should fit its journal easily, in order that the pressure at the bearing surface may be simply equal to the resultant of the forces applied to the shaft (including its weight), and may not be increased by means of any *grip* of the bush upon the journal. In other words, the journal should *bottom* in the bush. At the same time, there should be no such slackness as will admit of perceptible *shaking* of the journal in the bush.

CHAPTER II.

OF MARINE STEAM-ENGINES.

SECTION I.—MECHANICAL ACTION OF HEAT, ESPECIALLY THROUGH STEAM.

19. *General Explanations respecting Heat.*—The properties of that condition of bodies called *Heat*, to which it is necessary to refer in this Treatise, are the following:—

I. Heat is *transferrible* from one body to another; that is, a hotter body can heat a colder body by becoming less hot itself; and the tendencies to effect that transfer are capable of being compared together by means of a scale of quantities on which they depend, called *temperatures*.

II. The transfer of the condition of heat between two bodies tends to bring them to a state called that of *uniform temperature*, at which the transfer ceases.

III. The quantities called temperatures are accompanied in each body by certain conditions as to the relations between volume and elasticity; the general law being, that the hotter a body is, the less is its *elasticity of figure*, or tendency to preserve a definite form and arrangement of parts, and the greater its *elasticity of volume*; that is, its tendency, if solid or liquid, to preserve a definite volume, and if gaseous, to expand indefinitely.

IV. The condition of heat is a condition of *energy*; that is, of capacity to effect changes. One of those changes has already been mentioned under the head I., viz., the change in the

condition of heat of bodies which are unequally hot, tending to bring them to uniformity of temperature. Amongst other of those changes are changes of volume, changes of elasticity, chemical, electrical, and magnetic changes. The effort or tendency of heat to produce change of any kind depends on the temperature.

V. The condition of heat, considered as a kind of energy, is capable of being indirectly measured, so as to be expressed as a *quantity*, by means of one or other of the directly measurable effects which it produces.

VI. When the condition of heat is thus expressed as a quantity, it is found to be subject, like other forms of energy, to the law of *conservation*: that is, if in any system of bodies no heat is expended or produced through changes other than changes of temperature, then the total quantity of heat in the system cannot be changed by the mutual actions of the bodies; but what one body loses, another gains; and if there are changes other than changes of temperature, then if by those changes the total heat of the system is changed in amount, that change is compensated exactly by an opposite change in some other form of energy. For example, when mechanical work is done by means of heat (as in the steam-engine), a quantity of heat disappears bearing a fixed ratio to the work done; and when mechanical work is expended in producing heat (as in the friction of machinery),

the heat produced bears the same fixed ratio to the work so expended.

20. Temperatures—Thermometers—Elasticity of Perfect Gases.

—Two bodies are said to be at *equal temperatures*, or at the *same temperature*, when there is no tendency to the transfer of heat from either to the other.

Fixed temperatures, or *standard temperatures*, are temperatures identified by means of certain phenomena which occur at them.

The most important and useful of fixed temperatures are, that of the melting of ice, and that of the boiling of pure water under the average atmospheric pressure (called one atmosphere) of

14.7	lbs. on the square inch, or
2116.3	lbs on the square foot, or
29.922 inches (= 760 millimetres)	of a vertical column of mercury at the
	temperature of melting ice, or
10333	kilogrammes on the square metre.

The two *standard points* of the scale of temperatures (commonly called the *freezing point* and the *boiling point*) having been thus fixed, it is next requisite to express all other temperatures by means of a scale of *degrees*, and fractions of a degree; which scale is to be graduated according to the magnitude of some directly measurable quantity depending on temperature.

The quantity chosen for that purpose is the *elasticity*, or product of the pressure and volume, of a given mass of a perfect gas.

A *perfect gas* is a substance in such a condition, that the total pressure exerted by any number of portions of it, at a given temperature, against the sides of a vessel in which they are inclosed, is the sum of the pressures which each such portion would exert if inclosed in the vessel separately at the same temperature. Absolutely perfect gases are not found in nature; but every gas approximates more closely to the condition of a perfect gas the more it is heated and rarefied; and air is sufficiently near to the condition of a perfect gas for thermometric purposes.

All perfect gases increase in elasticity in the same proportion in rising from one given temperature to another; and in rising from the freezing point to the boiling point, the elasticity of a perfect gas increases in the ratio of

1:365 : 1, very nearly.

The thermometric scales most used are the Centigrade and Fahrenheit's. On the Centigrade scale, the interval from the freezing point to the boiling point is divided into 100 degrees, and the *zero* is at the freezing point. On Fahrenheit's scale, the same interval is divided into 180 degrees, and the *zero* is 32 of those degrees below the freezing point. Temperatures measured downwards from zero are marked by the negative sign, or *minus* (—).

The *absolute zero* is the temperature, known by theory only, at which the elasticity of a perfect gas would vanish.

Absolute temperatures are temperatures reckoned from the absolute zero. On the absolute scale, there are no negative temperatures.

The following is a general comparison of standard points and zeros on thermometric scales:—

	Common Scales.		Absolute Scales.	
	Cent.	Fahr.	Cent.	Fahr.
Water boiling under one } mean atmosphere,.....)	100°	212°	374°	673°·2
Melting ice,	0	32	274	493·2
Fahrenheit's zero,.....	—17·8	0	256·22	461·2
Absolute zero,.....	—274	—461·2	0	0

Absolute temperatures are the best to use in stating scientific principles and in calculations; but as the absolute zero is known approximately only (the estimates of its position by different authors varying to the extent of about a Centigrade degree), temperatures stated according to a common scale are the most convenient for observation and practice. In the present Treatise, when not otherwise specified, temperatures will be stated according to Fahrenheit's common scale, as being the most familiar to British engineers.

The following are rules for converting temperatures:—

I. Intervals of temperature:—

5 degrees Centigrade = 9 degrees Fahrenheit.

1 degree Centigrade = 1·8 degree Fahrenheit.

1 degree Fahrenheit = $\frac{5}{9}$ degree Centigrade.

II. Temp. Centigrade = $\frac{5}{9}$ (Temp. Fahrenheit — 32°).

III. Temp. Fahrenheit = 32 + 1·8 Temp. Centigrade.

IV. Absolute temperature = common temperature + absolute temperature of zero of common scale: that is, 274° Centigrade, or 461°·2 Fahrenheit.

For practical purposes, the scale of the mercurial thermometer may be taken as sensibly agreeing with that of the perfect gas thermometer.

21. *Elasticity of Gases.*—These three quantities for a given weight of a given substance in the perfectly gaseous state—its absolute temperature, its volume, and its absolute pressure*—are so connected, that if any two of them are given, the following principle enables the third to be calculated:—

I. The *product of the pressure and volume is proportional to the absolute temperature.*

The product of the pressure and volume at some standard temperature (as that of melting ice) is ascertained by experiment for each particular gas.

In such calculations, the pressure ought to be reduced to the same units of measure with the weight and volume; for example, the weight considered being one lb., and the volume expressed in cubic feet, the pressure ought to be expressed in *lbs. on the square foot*.

The following is a table of constants for different gases:—

	Elasticity, or product of Pressure and Volume, at the temperature of melting ice, in foot-lbs. to the lb.
Air.....	26214
Steam-gas (that is, perfectly gaseous steam),.....	42140
Oxygen,.....	23710
Nitrogen,.....	26990
Hydrogen,	378819
Carbonic acid (if perfectly gaseous),.....	17264
Do. (actual),.....	17145

EXAMPLE.—What is the elasticity of a pound of steam-gas at 320° Fahr.?

Absolute temperature, 320° + 461°·2 = 781°·2 Fahr.

$$42140 \times \frac{781 \cdot 2}{493 \cdot 2} = 42140 \times 1 \cdot 5839 = 66746 \text{ ft.-lbs.}$$

In order that steam of a given pressure may be sensibly in the perfectly gaseous state, it must be considerably *superheated*; that is, raised above the temperature at which water boils under the given pressure. *Saturated steam*—that is, steam as it comes from the water in the act of boiling—has less elasticity than steam-gas at the same temperature. The following rule gives a rough approximation, for practical purposes, to the elasticity of a pound of steam-gas, when the elasticity and pressure of a pound of saturated steam of the same temperature are known:—

II. To the elasticity of the saturated steam in foot-lbs. on the lb., add 38 times the square root of its pressure in lbs. on the

* *Absolute pressure* is the total intensity of the pressure exerted by an elastic substance. A pressure-gauge shows only the *difference* between the absolute pressure and the pressure of the atmosphere.

square foot (or 456 times the square root of its pressure in lbs. on the square inch).

EXAMPLE.—Given, pressure of saturated steam, 12940 lbs. on the square foot (= 89.86 lbs. on the square inch):

Volume of one lb., 4.816 cubic feet.

Then, elasticity of one lb. of saturated steam = $12940 \times 4.816 = \dots 62330$ ft.-lbs.

$88 \times \sqrt{12940} = 88 \times 113.8 = \dots 4324$ "

Elasticity of one lb. of steam-gas at the same temperature, nearly, 66654 "

The exact answer is 66746, showing an error of 96 foot-lbs., or about $\frac{1}{750}$ th part.

Suppose it is now required to find the temperature of that steam; then—

$\frac{66654}{42140} = 1.5817$; and

$493^{\circ}2 \times 1.5817 =$ absolute temperature, $\dots 780^{\circ}1$ Fahr.

Subtract, $\dots 461^{\circ}2$ "

Common temperature, $\dots 318^{\circ}9$ "

The exact answer is 320° Fahr., showing an error of a little more than one degree; which would be a large error for scientific purposes, but is small in practical questions relating to steam-engines.

22. *Expansion of Liquids and Solids—Fusion of Solids.*—The rate of expansion of every liquid and solid increases as the temperature becomes higher, and diminishes as the temperature becomes lower.

In the case of water, there is a temperature at which the rate of expansion disappears, and the volume of a given weight reaches a minimum. That temperature, according to the most trustworthy experiments, is $39^{\circ}1$ Fahr.

Between that temperature and 32° , the volume of a given weight of water *increases by cold*.

It is possible that a similar phenomenon may take place in other liquids; but it has not yet been observed in any liquid except water.

For rough calculations of the expansion of water, suited for most practical purposes, the following rule is sufficient:—Divide 500 by the absolute temperature in Fahrenheit's degrees, and the same absolute temperature by 500; the half sum of the quotients will be the ratio of the volume of the water at that temperature to its least volume. For the Centigrade scale, use 278 instead of 500.

The numbers which it is customary to give in tables of the *expansion of solids*, are the *rates of expansion of one dimension*, and are therefore respectively *one-third* of the corresponding rates of expansion in volume.

Solid thermometers are sometimes used, which indicate temperatures by showing the difference between the expansions of a pair of bars of two substances whose rates of expansion are different. When such thermometers are used to indicate temperatures higher than the boiling point of mercury under one atmosphere (about 676° Fahr.), they are called *Pyrometers*. In that case the exact value of their degrees is somewhat uncertain.

The following table shows the rates of expansion of some liquid and solid substances, when the temperature rises from that of melting ice to that of water boiling under one atmosphere:—

LIQUIDS.—(Expansion in Volume.)

Pure water,	0.04775
Sea water, ordinary,	0.05
Spirit of wine,	0.1112
Mercury,	0.018153
Oil, linseed and olive,	0.08

SOLIDS, METALLIC.—(Expansion in Length.)

Brass,	0.00216
Bronze,	0.00181
Copper,	0.00184
Cast-iron,	0.00111
Wrought-iron and steel,	0.00125 to 0.00114
Lead,	0.0029
Tin,	0.0022
Zinc,	0.00294

SOLIDS, NON-METALLIC.

Brick, common,	0.00355
" fire,	0.0005
Cement,	0.0014
Glass (average of different kinds),	0.0009
Slate,	0.00104
Deal timber, dry, lengthwise,	0.00043

The following are the melting points of a few of the more important substances. Those marked ? have been measured by the pyrometer:—

	Fahr.		Fahr.
Mercury,	-38°	Bismuth,	493°
Ice,	$+32^{\circ}$	Lead,	630°
Alloy—Tin 3, lead 5, bismuth 8,	210°	Zinc,	$700^{\circ}?$
about,	210°	Silver,	$1280^{\circ}?$
Sulphur,	228°	Brass,	$1869^{\circ}?$
Alloy—Tin 4, bismuth 5, lead 1,	246°	Copper,	$2548^{\circ}?$
Alloy—Tin 1, bismuth 1,	286°	Gold,	$2590^{\circ}?$
Alloy—Tin 3, lead 2,	334°	Cast-iron,	$3479^{\circ}?$
Alloy—Tin 2, bismuth 1,	334°	Wrought-iron, higher, but uncertain,	
Tin,	426°		

Ice, cast-iron, and several other substances, are more bulky when in the solid state, near the melting point, than they are when in the liquid state; as is shown by the solid material floating in the melted material.

23. *Pressure of Vapour—Evaporation—Boiling.*—The temperature at which a given fluid boils under a given pressure, is a fixed temperature. In order to explain that phenomenon, and the laws which it follows, it is necessary to describe the distinctions between the liquid and gaseous conditions, and the mode in which substances pass from the one to the other.

I. The *liquid* state is that condition of each internal part of a body, which consists in tending to preserve a definite volume, and resisting change of volume, and in offering no resistance to change of figure. The property of offering no resistance to change of figure, is common to the condition of *liquid* and *gas*, and constitutes the *fluid* condition. The liquid condition is distinguished from the gaseous by the property of tending to preserve a definite volume: a body in the gaseous condition tends to expand indefinitely. Rise of temperature increases the resistance of liquids to compression, and diminishes their cohesion. It is known of most liquids, and believed of all, that for each temperature of a given substance, there is a certain minimum pressure on its external surface, which is necessary to its existence in the liquid state, and under which the communication of additional heat to the liquid mass makes it *boil*, or emit bubbles of vapour from its interior. There is also reason to believe, that all liquids under all circumstances emit vapour from their surfaces, and are surrounded by an atmosphere of their own vapour.

II. *Vapour* is any substance in the gaseous condition, at the maximum of density consistent with that condition.

III. *Pressure and Density of Vapours.*—For each volatile substance at each temperature, there is a certain pressure which is at once the least pressure under which the substance can exist in the liquid or solid state, and the greatest pressure which it can sustain in the gaseous state, at the given temperature. That pressure is called the *pressure of saturation*, or the *pressure of*

vapour of the given substance at the given temperature; it is a function of (that is, a quantity depending on) the temperature; and the density of the vapour is a function of the pressure and the temperature. The relation between the pressure of vapour and the temperature, for various substances, has been the subject of many series of experiments, of which the latest and best are those of M. Regnault on steam, and on various other vapours. The result of such experiments is, that at temperatures which occur in practice, the pressure of vapour increases with the temperature at a rate which itself increases rapidly with the temperature. At very high temperatures, the pressure of each vapour appears to approximate to a limit which it cannot pass.*

If any vapour were a perfect gas, the volume of a given weight of it, at any given temperature, might be calculated by Rule I. of Article 21, supposing the volume of a given weight at some one temperature to have been ascertained by experiment; but no vapour is an absolutely perfect gas, and the heaviness of every vapour increases more rapidly with increase of pressure than that which would be given by the rule. That rule, however, is sufficiently near the truth for practical purposes when the heaviness of the vapour is below certain limits, as is the case with the vapours of most substances at the temperatures which usually occur in the atmosphere.

The following approximate rule serves, by the help of a table of logarithms, to calculate the volume of a pound of saturated steam at a given pressure:—

RULE A.—Reduce the absolute pressure to atmospheres (by dividing by 14·7, if it is stated in lbs. on the square inch). Take the 16th power of the 17th root of the pressure in atmospheres, and divide 26·36 by it, for the volume of a pound of steam in cubic feet.†

The following is Messrs. Fairbairn & Tate's Rule:—

RULE B.—From the absolute pressure in lbs. on the square inch, subtract 0·35; divide 389 by the remainder; to the quotient add 0·41: the sum will be the volume of one pound of steam in cubic feet.

EXAMPLE.—Pressure 29·4 — 0·35 = 29·05

389 ÷ 29·05 = 13·39

Add..... 0·41

Volume required, 13·8 cubic feet.

* The following formula for calculating the pressure p of vapour from the absolute temperature $t = T + 461^{\circ}\cdot 2$ Fahr. of the boiling point, was first given in the *Edinburgh Philosophical Journal* for July, 1849, and afterwards, with revised constants, in the *Philosophical Magazine*, December, 1854:—

$$\log p = A - \frac{B}{t} - \frac{C}{t^2}$$

The following is the inverse formula for calculating the absolute temperature of the boiling point from the pressure:—

$$t = 1 \div \left\{ \sqrt{\left(\frac{A - \log p}{C} + \frac{B^2}{4C^2} \right)} - \frac{B}{2C} \right\}$$

The following are a few values of the constants in the formula, for temperatures in degrees of Fahrenheit, and pressures in pounds on the square foot:—

Fluid.	A.	log. B.	log. C.	$\frac{B}{3C}$	$\frac{B^2}{4C^2}$
Water,	8·2591 ...	3·43642 ...	5·59873 ...	0·008441 ...	0·00001184
Ether,	7·5732 ...	3·31492 ...	5·21706 ...	0·006264 ...	0·00003924
Bisulphuret of carbon,	7·3438 ...	3·30728 ...	5·21839 ...	0·006136 ...	0·00003765
Mercury,	7·9691 ...	3·72284			

For inches of mercury at 32°, subtract from A,..... 1·8496

" lb. on the square inch, " A,..... 2·1584

For the Centigrade scale, subtract from log. B,..... 0·25527

log. C,..... 0·51054

also multiply $\frac{B}{2C}$ by 1·8, and $\frac{B^2}{4C^2}$ by 3·24

† That is, multiply the logarithm of the pressure in atmospheres by 16, and divide by 17; subtract the quotient from log. 26·36 = 1·4209 (four figures being enough); the sum will be the logarithm of the required volume.

EXAMPLE.—What is the volume of one lb. of steam, at the absolute pressure of 29·4 lbs. on the square inch = two atmospheres?

Log. pressure,..... 0·30103 $\times \frac{16}{17}$ = 0·2833

To be subtracted from log. 26·36..... 1·4209

Remainder,..... 1·1376

Being the log. of 13·78, volume required in cubic feet.

In Plate K, the relations between the pressure, volume, and temperature of saturated steam are shown by means of a curve, drawn through a network of two sets of equidistant parallel lines. Each vertical division represents a pound of absolute pressure on the square inch; each horizontal division, a cubic foot of volume to the pound weight of steam. Temperatures are marked at every fifth degree Centigrade, and every ninth degree of Fahrenheit.

IV. *Evaporation and Condensation*.—When the density of the vaporous atmosphere of a solid or liquid is diminished, either by the enlargement of the space in which the substance is contained, or by the removal of part of the vapour, whether by mechanical displacement (as when it is blown away by a current of air) or by condensation in an adjoining space, the solid or liquid evaporates until equilibrium is restored, by the restoration of the vapour to the density corresponding to the existing temperature. The same thing takes place when the molecular equilibrium is disturbed by communicating heat to the solid or liquid. When the density of the vaporous atmosphere is increased, either by the contraction of the space in which the substance is contained, or by the addition of vapour from another source, part of the vapour condenses until equilibrium is restored as before. The same thing takes place when the molecular equilibrium is disturbed by abstracting heat from the vapour. Evaporation is accompanied by the disappearance of heat, called the *Latent Heat of Evaporation*, and condensation by the reappearance of heat, according to laws to be stated farther on.

V. *Boiling*.—When the communication of heat to a liquid mass and the removal of the vapour are carried on continuously, so that the pressure throughout the mass of liquid is not greater than that of saturation for its temperature, evaporation takes place, not merely from the exposed surface of the liquid, but also from its interior: it gives out bubbles of vapour, and is said to *boil*. When the term *boiling point* of a fluid is used without qualification, it means the boiling point under the average atmospheric pressure of 14·7 lbs. on the square inch.

VI. *Superheated Vapour* means vapour which has been brought to a temperature higher than the boiling point corresponding to its pressure, so as to be in the condition of a permanent gas. Steam superheated about 20° or 30° is sensibly in the perfectly gaseous condition. See Article 21.

VII. *Resistance to Boiling—Brine*.—The presence in a liquid of a substance dissolved in it (as salt in water) resists ebullition, and raises the temperature at which the liquid boils, under a given pressure; but unless the dissolved substance enters into the composition of the vapour, the relation between the temperature and pressure of saturation of the vapour remains unchanged. The boiling point of saturated brine under one atmosphere is 226° Fahr., and that of weaker brine is higher than the boiling point of pure water by 1°·2 Fahr. for each $\frac{1}{32}$ of salt that the water contains. Average sea water contains $\frac{1}{32}$; and the brine in marine boilers is not suffered to contain more than from $\frac{3}{32}$ to $\frac{1}{32}$.

VIII. *Cloudy Vapour* is a condition of fluids in which the liquid floats in the air, or in its own vapour, in the form of innumerable small globules. The condition of cloud is one into which fluids pass from the state of vapour on being condensed by mingling with cold air. By heat, the globules of cloud are made to evaporate and disappear.

IX. Mixtures of Vapours and Gases.—The pressure exerted against the interior of a vessel by a given quantity of a perfect gas inclosed in it, is the sum of the pressures which any number of parts into which such quantity might be divided would exert separately, if each were inclosed in a vessel of the same bulk alone, at the same temperature. The same law is applicable to mixtures of gases of different kinds.

A second law of mixtures of gases and vapours is, that the presence of a foreign gaseous substance in contact with the surface of a solid or liquid, does not affect the density of the vapour of that solid or liquid, unless there is a tendency to chemical combination between the two substances, in which case the density of the vapour is slightly increased.

24. Quantities of Heat.—The condition of heat is measured as a quantity, and its amounts in different bodies and under different circumstances are compared, by means of the changes in some measurable phenomenon produced by its transfer or disappearance. The change most commonly used for that purpose is change of temperature. Heat employed in producing elevation of temperature is called *sensible heat*.

Equal differences of temperature in the same body correspond to equal quantities of heat in perfectly gaseous bodies alone. In bodies in other conditions, equal differences of temperature do not exactly correspond to equal quantities of heat.

For the purpose of expressing and comparing quantities of heat, it is convenient to adopt as an **UNIT OF HEAT OR THERMAL UNIT**, that quantity of heat which corresponds to some definite interval of temperature in a definite weight of a particular substance.

The thermal unit employed in Britain is *the quantity of heat which corresponds to an interval of one degree of Fahrenheit's scale in the temperature of one pound of pure liquid water, at and near its temperature of greatest density (39°·1 Fahrenheit)*.

The thermal unit employed in France (called *Calorie*) is *the quantity of heat which corresponds to an interval of one Centigrade degree in the temperature of one kilogramme of pure liquid water, at and near its temperature of greatest density (3°·94 Centigrade)*.

The following statement shows the mutual ratios of the British and French thermal units, with the logarithms of those ratios:—

	Ratios.	Logarithms.
British thermal units in a French thermal unit,	3·96832 ...	0·5986065
French thermal unit in a British thermal unit,	0·251996 ...	1·4013935

Other units in which quantities of heat can be expressed will be afterwards explained.

25. Specific Heat of Liquids and Solids.—The *specific heat* of a given substance means the quantity of heat, expressed in thermal units, which must be transferred to or from an unit of weight (such as a pound) of that substance, in order to raise or lower its temperature by one degree.

The specific heat of liquid water at and near its temperature of maximum density is *unity*; and the specific heat of any other substance, or of water itself at another part of the scale of temperatures, is the *ratio of the weight of water at or near 39°·1 Fahrenheit, which has its temperature altered one degree by the transfer of a given quantity of heat, to the weight of the other substance under consideration, which has its temperature altered one degree by the transfer of an equal quantity of heat*.

The specific heat of a substance is sometimes called its "*capacity for heat*."

SPECIFIC HEAT OF SOME LIQUIDS AND SOLIDS.

Water at 4° Cent. = 39° Fahr. nearly,	1·0000
" 54 " 129 "	1·0025
" 104 " 219 "	1·0100
" 154 " 309 "	1·0225
" 204 " 399 "	1·0400
" 254 " 489 "	1·0625
Mercury,	0·033
Ether,	0·517
Olive oil,	0·310
Copper,	0·095
Iron,	0·114
Lead,	0·031
Tin,	0·051
Brick, building stones, mortars, and cements,	from 0·2 to 0·22
Non-metallic materials and contents of a furnace,	" 0·2 to 0·22
Zinc,	0·093
Ice,	0·504
Sulphur,	0·203
Charcoal,	0·242
Coal and coke (average),	0·201
Silica (pure flint and sand),	0·191
Flint-glass,	0·19

26. Specific Heat of Gases.—The specific heat of a gas which is nearly in the perfectly gaseous state does not sensibly vary with density or with temperature; so that for such a gas, equal intervals of temperature correspond to equal quantities of heat on all parts of the thermometric scale.

The specific heat of a gas is different, according as it is maintained at a *constant volume*, or at a *constant pressure*, during the operation of changing its temperature; and the ratio which those two specific heats bear to each other is constant, or nearly so; the specific heat at constant pressure being the greater.

The reason is, that when the temperature of a gas is raised at constant volume, the heat produces no effect except raising the temperature; whereas when the pressure of the gas is maintained constant during the rise of temperature, the gas must be allowed to expand, and so to perform mechanical work; and in performing that work some heat disappears in addition to that employed in raising the temperature.

TABLE OF THE SPECIFIC HEAT OF SOME GASES.

	At constant Volume.	At constant Pressure.	Ratio.
Air,	0·169	0·238	1·408
Oxygen,	0·156	0·218	1·397
Hydrogen,	2·410	3·405	1·413
Nitrogen,	0·173	0·244	1·409
Steam,	0·370	0·480	1·297
Carbonic acid,	0·172	0·217	1·262

27. Latent Heat means, a quantity of heat which has *disappeared*; having been employed to produce some change other than elevation of temperature. By exactly reversing that change, the quantity of heat which had disappeared is reproduced.

The effects other than rise of temperature, produced by quantities of heat which disappear, can be used to measure and compare those quantities. Three of those effects are of importance in connection with the subject of the present Section:—Expansion, Fusion, and Evaporation.

I. LATENT HEAT OF EXPANSION.—Heat which disappears in causing the volume of a body to increase under a given pressure, has already been illustrated in the case of gases. (See Article 26.)

II. LATENT HEAT OF FUSION.—The following are examples in British thermal units per lb.:—

Substances.	Melting points. Fahr.	Latent heat of fusion.
Ice,	32°	140
Spermaceti,	56	148
Bees' wax,	140	175
Sulphur,	405	16·86
Tin,	426	500

III. LATENT HEAT OF EVAPORATION.—When a body passes from the solid or liquid to the gaseous state, its temperature

during the whole operation remains stationary at a certain *boiling point* (Article 23) depending on the pressure of the vapour produced; and in order to make the evaporation go on, a quantity of heat must be transferred to the substance evaporated, whose amount, for each unit of weight of the substance evaporated, depends on the temperature. That heat does not raise the temperature of the substance, but *disappears* in causing it to assume the gaseous state; and it is called the *latent heat of evaporation*.

When a body passes from the gaseous state to the liquid or solid state, its temperature remains stationary, during that operation, at the boiling point corresponding to the pressure of the vapour; a quantity of heat equal to the latent heat of evaporation at that temperature is produced in the body; and in order that the operation of condensation may go on, that heat must be transferred from the body condensed to some other body.

The following are examples of the latent heat of evaporation in British thermal units, of one pound of certain substances, when the pressure of the vapour is *one atmosphere* of 14·7 lbs. on the square inch:—

Substances.	Boiling point under one atm. Fahr.	Latent heat in British units.
Water,.....	212°·0	966
Alcohol,.....	172·2	364
Æther,.....	95·0	163
Bisulphuret of carbon,.....	114·8	156

The latent heat of evaporation of water, or **LATENT HEAT OF STEAM**, as it is commonly called, diminishes nearly at the rate of *seven-tenths of an unit of heat for each degree of rise of the boiling point*. Hence the following

RULE.—Take the difference between the actual boiling point and the boiling point under one atmosphere (212° Fahr., or 100° Cent.); multiply it by 0·7; subtract the product from, or add it to, the latent heat under one atmosphere (966 British or 537 French units), according as the actual boiling point is above or below the boiling point under one atmosphere: the result will be the latent heat of steam (one pound or one kilogramme as the case may be) at the actual boiling point.

This rule is approximative only, but is near enough for practical purposes.

EXAMPLES.—Required the latent heat of one pound of steam at 104° Fahr. and at 320° Fahr. respectively:

Actual boiling points,.....	104°	320°
Boiling point under one atmosphere,.....	212	212
Differences,.....	—108	+108
Multiply by,.....	0·7	0·7
Products (to the nearest unit only),.....	+76	—76
Latent heat under one atmosphere,.....	966	966
Latent heat required,.....	1042	890

28. *Total Heat of Evaporation* (called, in the case of water, the **TOTAL HEAT OF STEAM**), means the quantity of heat required to raise the temperature of an unit of weight of a liquid from a fixed temperature to a given boiling point, and then to evaporate it at that boiling point. It is described as “the total heat from the fixed temperature, at the given boiling point.” It is the sum of two parts, viz.—the *sensible heat*, being the product of the specific heat of the liquid into the rise of temperature; and the *latent heat* at the given boiling point.

In formulæ and tables relating to the total heat of steam, it

is usual to take for the fixed temperature, that of melting ice (32° Fahr., or 0° Cent.). The total heat of steam, from 32° Fahr., at 212° Fahr., is very nearly as follows, in British units per lb.,

$$\text{Sensible heat } 180 + \text{latent heat } 966 = 1146, \text{ total heat;}$$

and in French units per kilogramme,

$$\text{Sensible heat } 100 + \text{latent heat } 537 = 637, \text{ total heat.}$$

The total heat of steam *increases* nearly at the rate of *three-tenths of an unit of heat for each degree of elevation of the boiling point*, and *diminishes* nearly at the rate of *an unit of heat for each degree of elevation of the temperature of the feed-water*. Hence the following

RULE.—To find the total heat of an unit of weight of steam, from a given temperature of feed-water, at a given boiling point. To the constant 1146 in British measures (or 637 in French measures), add $\frac{3}{10}$ of the elevation of the boiling point above the boiling point under one atmosphere (212° Fahr., or 100° Cent.); then from the sum subtract the elevation of the temperature of the feed-water above that of melting ice (32° Fahr., or 0° Cent.).

When the boiling point is below 212° Fahr., or 100° Cent., the first operation is a subtraction instead of an addition.

EXAMPLE.—Required the total heat of one pound of steam, in British measures, from 104° Fahr. (temperature of feed), at 320° Fahr. (boiling point):

Constant,.....	1146
Add $\frac{3}{10} (320^\circ - 212^\circ) = \frac{3}{10} \times 108^\circ =$	32·4
Sum,.....	1178·4
Subtract $104^\circ - 32^\circ =$	72·0
Total heat required, to the nearest unit,.....	1106

29. *Measurement of Heat by Evaporation.*—The heat produced by the combustion of a given weight of fuel (of which examples will be given in Chapter III.), is usually ascertained by finding what weight of water it evaporates. In such experiments, it is essential to the obtaining of accurate results that the temperature of the feed-water and the temperature of evaporation should both be ascertained, and the total heat per pound of water computed. That total heat, in British units, being divided by 966, the latent heat of evaporation of a pound of water at 212° Fahr., gives a *multiplier*, by which the weight of water actually evaporated by each pound of fuel is to be multiplied, to reduce it to the *equivalent evaporation from and at 212° Fahr.*; that is, *the weight of water which would have been evaporated by each pound of fuel, had the water been both fed and evaporated at the boiling point corresponding to the mean atmospheric pressure*.

The weight of water so calculated is called the *evaporative power* of the fuel. To state it is, in fact, to employ a peculiar thermal unit, viz., the latent heat of evaporation of one pound of water at 212° Fahr., which is 966 times greater than the ordinary British thermal unit. To exemplify the reduction above described, let the water be supplied to the boiler at 104° Fahr., and evaporated at 320° Fahr. Then the total heat of evaporation in common British units per pound of steam is 1106; and the factor of evaporation is $\frac{1106}{966} = 1·145$

The following is a table of factors of evaporation,* correct

* Extracted from “A Manual of the Steam Engine and other Prime Movers.”

to the second place of decimals; that is, there is no error greater than '005:—

TABLE OF FACTORS OF EVAPORATION.

Boiling Point, Fahr.	Temperature of Feed-water, Fahr.										
	32°	50°	68°	86°	104°	122°	140°	158°	176°	194°	212°
212°.....	1.19	1.17	1.15	1.13	1.11	1.10	1.08	1.06	1.04	1.02	1.00
230°.....	1.20	1.18	1.16	1.14	1.12	1.10	1.08	1.06	1.04	1.02	1.01
248°.....	1.20	1.18	1.16	1.14	1.13	1.11	1.09	1.07	1.05	1.03	1.01
266°.....	1.21	1.19	1.17	1.15	1.13	1.11	1.09	1.07	1.06	1.04	1.02
284°.....	1.21	1.20	1.18	1.16	1.14	1.12	1.10	1.08	1.06	1.04	1.02
302°.....	1.22	1.20	1.18	1.16	1.14	1.12	1.11	1.09	1.07	1.05	1.03
320°.....	1.22	1.21	1.19	1.17	1.15	1.13	1.11	1.09	1.07	1.05	1.03
338°.....	1.23	1.21	1.19	1.17	1.15	1.14	1.12	1.10	1.08	1.06	1.04
356°.....	1.23	1.22	1.20	1.18	1.16	1.14	1.12	1.10	1.08	1.06	1.04
374°.....	1.24	1.22	1.20	1.18	1.17	1.15	1.13	1.11	1.09	1.07	1.05
392°.....	1.24	1.23	1.21	1.19	1.17	1.15	1.13	1.11	1.09	1.07	1.06
410°.....	1.25	1.23	1.22	1.20	1.18	1.16	1.14	1.12	1.10	1.08	1.06
428°.....	1.25	1.24	1.22	1.20	1.18	1.16	1.14	1.12	1.11	1.09	1.07

29 A. *The Total Heat of Gasefication*, or quantity of heat required to convert a substance from the liquid or solid state at a fixed temperature, to the perfectly gaseous state at a given temperature, the operation being performed under a constant pressure, increases with the latter temperature at a rate equal to the specific heat of the gas under constant pressure; that is, for steam-gas 0.48, nearly. The constant part of the total heat of gasefication of steam-gas has not yet been ascertained experimentally; but if we assume that from and at the temperature of melting ice, the total heat of gasefication of steam-gas is the same with the total heat of steam (viz., 1092 in British measures, or 607 in French measures), the total heat of gasefication from the temperature of melting ice, at any higher temperature, is to be found by adding to the constant just mentioned $\frac{4.8}{100}$ of the elevation of the latter temperature above that of melting ice. The saving of heat through the feed-water being hotter than melting ice is the same as for ordinary steam.

EXAMPLE.—Required the total heat of gasefication, in British measures, of one pound of steam-gas, from 104° Fahr. (feed), at 320° Fahr.:

Constant.....	1092
Add $0.48 \times (320^\circ - 32^\circ) = 0.48 \times 288 =$	138
Sum.....	1230
Subtract $104^\circ - 32^\circ =$	72
Total heat of gasefication required.....	1158

The preceding rule must be regarded for the present as a provisional approximation only, because of the uncertainty of the value of the constant.

30. *The Two Laws of Thermodynamics*.—FIRST LAW.—*Heat and mechanical energy are mutually convertible; and heat requires for its production, and produces by its disappearance, mechanical energy in the proportion of 772 foot-pounds for each British unit of heat.*

The quantity just mentioned is called the *mechanical equivalent of an unit of heat*. It is also often called "*Joule's equivalent*," and denoted by the symbol J, in honour of Dr. Joule, who was the first to determine its value *exactly*. His first approximate determination of this quantity was published in 1843, a little after that of Mayer; his best set of experiments, from which the accepted value 772 is deduced, may be consulted in the Philosophical Transactions for 1850.

That value has been verified by many other researches; of which the most satisfactory to practical men must be those of Mr. G. A. Hirn, of Colmar, who made several careful measurements of the work done by the steam in the cylinder of a steam-

engine, of the whole heat expended on that steam, and of the heat carried off by the exhaust steam; and having compared the excess of the former quantity of heat above the latter (which excess disappears in the working of the engine) with the indicated work, found that their proportion to each other never deviated by more than two per cent. from Dr. Joule's value of the equivalent.^o

The following are the values of Joule's equivalent for different thermometric scales, and in French and British units:—

		J.
One British thermal unit, or degree of Fahrenheit in a lb. of water.....	772 foot-lbs.	
One Centigrade degree in a lb. of water.....	1389.6 "	
(or very nearly 1390).		
One French thermal unit, or Centigrade degree in a kilogramme of water.....	423.55 kilogrammètres.	
(or very nearly 424).		

All quantities of heat, such as the *specific heat* of any substance, or the *latent heat* corresponding to any physical effect, or any other of the quantities of heat treated of in the preceding Articles, may be expressed *dynamically*, that is, in units of work, by multiplying their values in ordinary units of heat by Joule's equivalent. For example—

Latent heat of evaporation of 1 lb. of water, from and at 212° = $966 \times 772 =$	745,752 foot-lbs.
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SECOND LAW.—*If the absolute temperature of any uniformly hot substance be divided into any number of equal parts, the effects of those parts in causing work to be performed are equal.*

To understand how this law is applied, it must be observed that when any substance undergoes a change of dimensions through the action of heat upon it, the mechanical equivalent of the heat which disappears consists of two parts—the *external work*, performed in overcoming some external resistance, as when steam, in expanding, lifts a loaded piston; and the *internal work*, performed in overcoming the attractions or other forces exerted by the particles of the substance itself upon each other. In sensibly perfect gases alone, the internal work is insensible, and the heat which disappears equivalent to the external work alone. In other substances, the internal work is sometimes by far the greater part of the whole work. For example, when a pound of water is evaporated at 212° Fahr., the mechanical equivalent of the heat which disappears is 745,752 foot-pounds; from which subtracting the external work, being the product of the pressure and increase of volume in evaporating, 55,783 foot-pounds, the internal work done in overcoming the cohesion of the liquid water is found to be 689,969 foot-lbs.

The internal work is not directly measurable in any case; but the second law of thermodynamics enables the whole work, external and internal, equivalent to the whole heat which disappears, to be calculated from the external work by the following method:—

Conceive the absolute temperature at which the given change of dimensions takes place to be varied by an indefinitely small fraction of itself, and find the corresponding variation in the external work done; divide the latter variation by the variation of temperature, so as to obtain the rate at which the external work done varies with temperature (this is called the "metamorphic function"); multiply that rate by the absolute temperature: the product will be the heat which disappears.

^o "Théorie Mécanique de la Chaleur," par G. A. Hirn. Second edition; Paris, 1865; pp 95-46.

EXAMPLE I.—To calculate the mechanical equivalent of the latent heat of one lb. of steam at the pressure of one atmosphere:

Rate of variation of pressure with temperature, in lbs. } on the square foot, per degree Centigrade,	75.695
x excess of volume of one lb. of steam in cubic feet } (26.36) above volume of one lb. of water (0.017),	26.343
Product, being the variation of external work for the } same change of volume, in foot-lbs. per degree Cent.,	1994.033
x absolute temperature (Centigrade),	374
Product, being the mechanical equivalent of the latent } heat of evaporation, in foot-lbs.,	745,768, very nearly.

The preceding example is given to illustrate the direct application of the rule; but the more frequent use to which the rule has been put is its inverse application—viz., the heat which disappears in evaporation being known, and also the law of variation of the pressure with the boiling point, to find the increase of volume in evaporating. The following example illustrates this:—

EXAMPLE II.—Given the mechanical equivalent of the latent heat of evaporation of one lb. of steam under the pressure of one atmosphere, as below:

Divide by absolute temperature, 374° Cent.,) 745,752 foot-lbs.	
Divide by rate of variation of pres- } sure per degree Centigrade in } 75.695) 1994.033 { Rate of variation of lbs. on the square foot,	external work per Centigrade degree.
Increase of volume of 1 lb. of water in evaporating, 26.343 cubic feet.	
Add volume of liquid water,	0.017 "
Volume of one lb. of steam at one atmosphere, ...	26.36 "

Such is the process by which the volumes of steam and other vapours were computed from the latent heat before they were ascertained by experiment.

31. *Efficiency of Engines in General.*—If the number of British thermal units produced by the combustion of one pound of a given kind of fuel, be multiplied by Joule's equivalent, 772 foot-pounds, the result is the *total heat of combustion* of the fuel in question, expressed in foot-pounds. For different kinds of fuel that quantity, in round numbers, ranges between 5,000,000 and 12,000,000 foot-pounds. That total heat is expended, in any given engine, in producing the following effects, whose sum is equal to the heat so expended:—

I. The *waste heat of the furnace*, being from 0.1 to 0.6 of the total heat, according to the construction of the furnace, and the skill with which the combustion is regulated.

II. The *necessarily rejected heat of the engine*: that is, the excess of the whole heat communicated to the working fluid by each pound of fuel burned, above the portion of that heat which permanently disappears, being replaced by mechanical energy.

III. The *heat wasted by the engine*, whether by conduction or by non-fulfilment of the conditions of maximum efficiency.

IV. The *useless work of the engine*, employed in overcoming friction and other prejudicial resistances.

V. The *useful work*. The efficiency of a heat engine is improved by diminishing as far as possible the first four of those effects, so as to increase the fifth.

It appears, then, that the efficiency of a marine steam-engine is the product of four factors, viz.:—I., The *efficiency of the furnace*, being the ratio which the heat transferred to the steam bears to the total heat of combustion; II., The *efficiency of the steam*, being the fraction of the heat received by it which is transformed into mechanical energy; III., The *efficiency of the mechanism*, being the fraction of that energy which is available for driving the propelling instrument; and IV., The *efficiency of*

the *propeller*, being the fraction of the last quantity of energy which is used in driving the vessel.

The first of those factors—the efficiency of the furnace—will be considered in Chapter III; the second—the efficiency of the fluid—is the special subject of the present section; the third will be considered in a subsequent Section; the fourth (and in part the third also) has been treated of in the preceding Chapter.

32. *Action of the Cylinder and Piston.*—The part of a heat-engine in which the fluid performs work consists essentially of an inclosed space whose volume is capable of being alternately enlarged and contracted, by the motion of one of its boundaries. In all engines that are extensively used in practice, the inclosed space is of a cylindrical form; and it is called the *CYLINDER*, even in those exceptional engines in which it has some other figure. Its moveable boundary is called the *PISTON*, and is usually a cylindrical disc fitting the cylinder, in which it moves to and fro in a straight line. In some exceptional engines the piston has other forms; but its action always is to increase and diminish alternately the volume of a certain inclosed space.

The steam, while it is entering the cylinder and expanding, drives the piston before it, and exerts on the piston an amount of energy equal to the product of the volume swept through by the piston into the mean intensity of the pressure of the steam. That operation is the *forward stroke*.

During the *return stroke*, or *backward stroke*, the piston drives the steam before it, and either expels it from the cylinder, or compresses it, or expels part and compresses part; and in so doing the piston exerts energy upon the steam to an amount equal to the product of the volume swept through by the piston into the mean intensity of the pressure of the steam, which is now called *back pressure*.

The excess of the energy exerted by the steam on the piston during the forward stroke above the energy exerted by the piston on the steam during the return stroke, is the *effective energy* exerted by the steam on the piston during one *complete stroke*, or *revolution*, consisting of a forward stroke and a return stroke, and is equal to the *work performed* by the piston in overcoming resistance other than the back pressure of the steam; and the amount of that work in some definite time, as a second, a minute, or an hour, is the *INDICATED POWER* of the engine.

It is to be borne in mind in such calculations that the spaces swept through by the piston, and the intensities of the pressure, must be stated in such units that the product of a space into the intensity of a pressure shall give a quantity of work. Thus, for quantities of work in *foot-pounds*—

UNIT OF PRESSURE.	UNIT OF SPACE.
One lb. on the square foot.	One cubic foot.
One lb. on the square inch.	A prism a foot long and an inch square = $\frac{1}{144}$ cubic foot;

and for quantities of work in *kilogrammètres*—

UNIT OF PRESSURE.	UNIT OF SPACE.
One kilogramme on the square metre,	One cubic metre.

Almost all marine steam-engines are *double-acting*; that is to say, the steam acts alternately on the two sides of the piston. In such engines the quantities of effective energy exerted by the steam on the two sides of the piston are to be found separately, and added together.

33. The *Indicator* consists essentially of a small cylinder communicating with the engine-cylinder, and having a piston and

a spring, for measuring the difference between the pressure of the steam and the atmospheric pressure: a pencil accompanying the movements of the small piston; and a card, or piece of paper wrapped round a small drum, which is so connected by cords and pulleys with some convenient part of the mechanism of the steam-engine, that its movements accurately keep time with, and represent on a small scale, the movements of the engine-piston. When a cock which connects the engine-cylinder with the indicator-cylinder is closed, the pencil traces on the



card a line, which, when the card is spread out flat, becomes the straight line, A A (Fig. 1). This is called the *atmospheric line*. After the cock is opened, the pencil, moving up and down with the variations of the pressure of the steam, traces on the card during each complete or double stroke a curve such as BCDEB. The ordinates drawn to that curve from any point in the atmospheric line, A A, such as HK and HG, indicate the differences between the pressure of the steam and the atmospheric pressure at the corresponding point of the motion of the piston. The ordinates of the part BCDE represent the pressures of the steam during the forward stroke, when it is driving the piston; those of the part EB represent the pressures of the steam when the piston is expelling it from the cylinder.

To found exact investigations on the indicator-diagrams of steam-engines, the atmospheric pressure at the time of the experiment ought to be ascertained by means of a barometer: but this is generally omitted; in which case the atmospheric pressure may be assumed at its mean value, being 14.7 lbs. on the square inch, or 2116.3 lbs. on the square foot, at and near the level of the sea.

Let $\overline{AO} = \overline{HF}$ be ordinates representing the pressure of the atmosphere. Then OFV, parallel to A A, is the *absolute* or *true* zero line of the diagram, corresponding to *no* pressure; and ordinates drawn to the curve from that line represent the absolute intensities of the pressure of the steam. Let OB and LE be ordinates touching the ends of the diagram. Then OL represents the *volume* swept by the piston at each single stroke;

The area OBCDELO represents the energy exerted by the steam on the piston during the forward stroke;

The area OBEL represents the work lost in expelling the steam during the return stroke;

The area BCDEB, being the difference of the above areas, represents the *effective work* of the steam on the piston during the complete stroke.

Those areas can be found by the methods explained in Articles 19 and 21 of Division I., pages 11 to 13. The number of intervals into which the length of the diagram is usually divided is ten; and the "trapezoidal rule" is considered sufficiently accurate in most cases. The planimeter is very useful for measuring indicator-diagrams.

The *mean forward pressure*, the *mean back pressure*, and the *mean effective pressure*, are found by dividing those three areas respectively by the volume represented by OL. (See Division I., Article 20, page 13.)

In a *Double-acting Engine* two indicators should be used at the same time, communicating respectively with the two ends of the cylinder. Thus a pair of diagrams will be obtained, one

representing the action of the steam on each face of the piston. The mean effective pressure is to be found for each diagram separately, and then, if the areas of the two faces of the piston are sensibly equal, *the mean of those two results* is to be taken as the *general mean effective pressure*; which being multiplied by the area of the piston, the length of stroke, and *twice* the number of double strokes or revolutions in a minute, gives the indicated power per minute, in units of work.

If the two faces of the piston are sensibly of unequal areas (as in "trunk engines"), the indicated power is to be computed separately for each face, and the results added together.

If there are two or more cylinders, the quantities of power indicated by their respective diagrams are to be added together. The indicated power in foot-pounds of work per minute being divided by 33,000, gives the *INDICATED HORSE-POWER*. (See the First Division, Article 66, page 28, bottom of the second column.)

The following is an example from a double-cylinder, double-acting engine, calculated by the trapezoidal rule:—

BREADTHS OF DIAGRAMS, MEASURED BY A SCALE REPRESENTING POUNDS ON THE SQUARE INCH.

Number of Intervals, 10.

	First Cylinder.		Second Cylinder.	
	Top.	Bottom.	Top.	Bottom.
First breadth,.....	27	36	16.0	12.4
Last breadth,.....	13	12	2.0	3.8
Sum,	40	48	18.0	16.2
Half sum,	20	24	9.0	8.1
Nine intermediate breadths,	88	97	10.5	10.8
	91	96	8.5	9.0
	91	84	7.5	8.0
	64	64	7.0	7.1
	57	57	6.6	6.7
	58	46	6.2	6.0
	42	40	6.0	5.6
	35	32	5.1	5.4
	22	22	4.5	5.0
Sum,	558	562	70.9	71.7
Sum ÷ 10 = mean effective pressure,	55.8	56.2	7.09	7.17
Mean of top and bottom,	56.0	...	7.13	...
× Area of piston, in square inches,	345	...	1380	...
Mean effort, in lbs.,	19320	...	9839.4	...
× Stroke, in feet, $2\frac{1}{2} \times$ revolutions } per minute, $52\frac{1}{2} \times 2 =$	262.5	...	262.5	...
Indicated power, in ft.-lbs. per minute,	5071500	...	2582842.5	...
Total indicated power, in foot-lbs. per minute,	7654342.5		...	
÷ 33000 = indicated horse-power,	232		...	

The *inertia* of the moving parts of the indicator causes oscillations of its piston above and below the curve which would accurately represent the pressures. In order that the errors arising from this cause may be as small as possible, it is advisable that the spring of the indicator should be stiff, and its moving parts light. This usually involves the use of a small scale of pressures; but the ordinates representing pressures may be magnified, either by mechanism (as in Richards' Indicator), or by using a magnifying glass in taking the measurements.

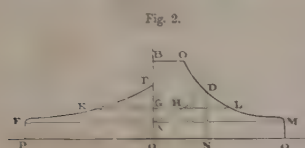
The *friction* of the moving parts of the indicator tends on the whole to make the indicated power and indicated mean effective pressure less than the truth, but to what extent is uncertain.

Every indicator should have the accuracy of the graduation of its scale of pressures frequently tested by comparison with a

standard pressure gauge; and great care should be taken to make the movements of the drum accurately correspond with those of the engine-piston.

The communication between the indicator and the engine-cylinder should be as direct and free as possible. Long narrow tubes are certain to cause errors.

34. Double-cylinder Engines—Combination of Diagrams.—In a double-cylinder engine, the steam performs its work in two cylinders, a smaller and a larger, which at certain periods communicate with each other. In some cases the functions of two cylinders are performed by the two ends of one cylinder. The details of such engines will be exemplified in a future Section; the object of the present Article being to show how the indicator-diagrams of work obtained from a double-cylinder engine are to be combined, so as to produce the diagram that would have



been obtained had the fluid performed the same work by going through the same series of changes of pressure and volume in one cylinder. The steam is first admitted from the boiler into the smaller cylinder, until it fills a certain volume, represented by BC in Fig. 2; the absolute pressure is represented by the height of BC above the zero line, POQ . The admission of the steam is then cut off, and it expands in the smaller cylinder with a pressure gradually diminishing, as shown by the ordinates of the curve CD . DN being let fall perpendicular to OQ , ON represents the whole space swept through by the piston of the smaller cylinder during its forward stroke. At the end of that stroke, a communication is opened between the smaller and the larger cylinder; and the forward stroke of the piston of the larger cylinder takes place at the same time with the return stroke of the piston of the smaller cylinder. During this process, the steam is driven before the piston of the smaller cylinder, and drives the piston of the larger cylinder; it exerts more energy on the latter piston than it receives from the former, because the piston of the larger cylinder sweeps through the greater space; and the difference between those quantities of energy is added to the energy formerly exerted by the steam on the piston of the smaller cylinder. This part of the action of the steam is represented by the curves DA and EF : the ordinates of DA representing the backward pressures exerted by the steam in the smaller cylinder, and the ordinates of EF , the forward pressures exerted by it at the same time in the larger cylinder. OP represents the space swept through by the piston of the larger cylinder, on the same scale with that according to which ON represents the corresponding space for the smaller cylinder.

The next operation is to shut the communication between the two cylinders, and open the exhaust port of the larger cylinder, and the admission port of the smaller. Then takes place the return stroke of the larger cylinder, during which the steam is expelled, exerting a back pressure represented by the ordinates of FA ; while at the same time a new portion of steam is admitted into the smaller cylinder, and expanded as before, during a new forward stroke of that cylinder.

Thus are produced the two indicator-diagrams, $BCDAB$ for the smaller cylinder, and $EFAE$ for the larger, and the sum of their areas represents the energy exerted on the pistons by

the quantity of steam which is expended at one stroke. When two such diagrams are taken by indicators, for the sole purpose of computing the power of an actual engine, they may be drawn on the same or on different scales, and the quantities of work indicated by them may be computed independently, and then added together. Of this a detailed example has already been given in the preceding Article.

But if the diagrams are to be used for the purpose of examining into the relations between heat expended and work performed, it is best to combine them into one diagram, in the following manner:—

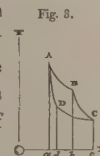
Draw any straight line KGH parallel to POQ , and intersecting both diagrams. Produce that line, and lay off upon it $HL = KG$.

Then $GL = GH + KG$ represents the total volume occupied by the steam, partly in the smaller and partly in the larger cylinder, when its absolute pressure is represented by OG ; and L is a point in the indicator-diagram which would have been described had the whole action of the steam taken place in the larger cylinder only.

By drawing a sufficient number of parallel lines, such as KL , and laying off the proper distances on them, as above, any number of points such as L may be found, so as to complete the combined diagram, $BCDLMAB$, whose length, $OQ = OP$, represents the volume swept through by the piston of the larger cylinder; and this diagram may be reasoned about as if it represented the action of the steam in the larger cylinder alone.

It is to be observed, then, as a general principle, that the energy exerted by a given portion of a fluid, during a given series of changes of pressure and volume, depends on that series of changes, and not on the number and arrangement of the cylinders in which those changes are undergone.

35. An Elementary Heat-Engine is one in which the reception of heat by the fluid takes place wholly at one absolute temperature, and its rejection wholly at another absolute temperature. Consequently, in such an engine, the change between those two limiting temperatures must be made entirely by compression and expansion of the fluid. The action of such an engine, during one stroke, consists of four operations, represented by the four sides of the diagram, $ABCD$, Fig. 3, as follows:—



- A B, expansion of the fluid at the higher limit of temperature;
- B C, further expansion, without reception or emission of heat, till the temperature falls to the lower limit;
- C D, compression of the fluid at the lower limit of temperature;
- D A, further compression, without reception or emission of heat, till the temperature rises again to the higher limit.

By a process which it is unnecessary here to give in detail, the second law of thermodynamics is shown to lead to the following LAW OF THE EFFICIENCY OF ELEMENTARY HEAT-ENGINES, viz.—That the heat transformed into mechanical energy is to the whole heat received by the fluid as the range of temperature is to the absolute temperature at which heat is received.

Between given limits of temperature, the efficiency of a heat-engine is the greatest possible, when the whole reception of heat takes place at the higher limit, and the whole rejection of heat at the lower; that is to say, when the engine is an elementary engine; and the efficiency of the fluid in such an engine is independent of the nature of the fluid employed: hence an

elementary heat-engine is also a *heat-engine of greatest efficiency* between given limits of temperature.

EXAMPLE.—What is the greatest possible efficiency of a heat-engine working between the limits 140° Cent. = 284° Fahr., and 40° Cent. = 104° Fahr.?

Absolute temperature, higher limit, $140 + 274 = \dots\dots 414^\circ$ Cent.
 " lower limit, $40 + 274 = \dots\dots 314^\circ$ "

Difference, or range of temperature, $\dots\dots\dots 100^\circ$ "

Greatest possible efficiency, $\frac{100}{414} = 0.2415$

That is to say, in a steam-engine supplied with steam from the boiler at 284° Fahr., corresponding to the absolute pressure of 52.52 lbs. on the square inch, and expelling the steam into the condenser at 104° Fahr., corresponding to the absolute pressure of 1.06 lbs. on the square inch, it is impossible to convert into mechanical energy more than 2415 out of every 10,000 units of heat expended on the steam; nor is it possible to prevent 7585 at least of those units from going with the exhaust steam to the condenser.

It is impracticable, in any actual steam-engine, to fulfil completely the conditions of greatest efficiency. In particular, it is impracticable to heat the feed-water from the lower to the higher limit of temperature wholly by the compression of part of the steam; and it is also impracticable to carry on the expansion of the steam until its temperature falls to that of the condenser. Further, it is necessary in actual engines to supply the steam with some heat during its expansion, in order that it may not partially liquefy in the cylinder. Hence the greatest efficiency of an elementary heat-engine, working between a given pair of temperatures, represents a limit which an actual engine may be made to approach, but which it cannot attain.

36. *Efficiency of the Steam in Existing Engines.*—In order to compute the efficiency of the steam in an existing engine, two quantities must first be determined—the indicated work of some definite quantity of steam, such as a pound, or a cylinder-full; and the heat expended upon that quantity of steam. The former of those quantities is computed from measurements of the indicator-diagram, as explained in Articles 33 and 34. The latter is to be found from the indicator-diagram also, by the aid of rules deduced from the principles of thermodynamics.

In engines which carry expansive working to any considerable extent, the total heat of the steam used, from the temperature of the feed and at the temperature of admission into the cylinder, forms by no means the whole expenditure of heat upon the steam; for a considerable additional quantity of heat must be communicated to the steam either while it is in the cylinder, or immediately before its admission, in order that liquefaction in the cylinder may be prevented, and the economy properly due to expansive working realized.

When the indicated work of the steam, and the expenditure of heat upon it, are expressed exactly, by formulæ deduced from the principles of thermodynamics, they appear in the shape of functions of the temperatures of the steam during its admission, at the end of its expansion, and during the exhaust, and of the temperature of the feed-water; and those functions are in general of a complex kind.* The finding of those temperatures, moreover, is a somewhat troublesome process. It is desirable, therefore, that for practical purposes there should be simple

rules for calculating the efficiency of the steam from pressures and volumes only, without considering temperatures. Such rules are special rules for steam only, and not applicable to other fluids; and they are not capable of the precision of the more exact formulæ; but their errors are unimportant in practice, so long as their application is confined within the ordinary limits of pressure.

In Fig. 4, let AFGHBKA represent the indicator-diagram of any steam-engine; F being the point of admission (marking where the steam is first introduced into the cylinder), G that of cut-off (marking where the admission stops and the expansion begins), B the point of release (marking where the exhaust-port is first opened to let the steam escape), H the end of the forward stroke, and K the point where cushioning or compression, if any, begins.

Let the horizontal line through C be the zero line of absolute pressures, so that heights above that line represent absolute pressures of the steam; BC, for example, being the absolute pressure at the instant of release.

Through B draw BA parallel to OC; and, if necessary, set back the point, A, so as to allow for clearance (that is, for steam-space in the ports and at the ends of the cylinder beyond the ends of the stroke), in order that the length, AB, may represent the whole volume of steam contained in the cylinder and ports at the instant of release. From A let fall the perpendicular, AO, upon the zero line.

Then if we calculate, in a series of particular cases, by the exact formulæ of thermodynamics, a quantity which may be called the *heat of release*, consisting of the total heat, sensible and latent, of the volume of steam, AB, at the absolute pressure, CB, together with the quantity of heat which that steam would carry off from the cylinder and valve-ports, supposing it to expand down to the back pressure without liquefaction, that quantity is found to be given approximately, to the accuracy of about one per cent., by the following rule:—

RULE FOR THE HEAT OF RELEASE.

I. Multiply the product (in foot-lbs.) of the absolute pressure and volume of the steam at the point of release by 16 for a condensing engine, or by 15 for a non-condensing engine: the result will be the mechanical equivalent of the heat of release, nearly.

This may, if desired, be reduced to ordinary thermal units by dividing by Joule's equivalent. (See Article 30.)

To represent the preceding rule graphically, in Fig. 1, produce AB to D, making AD = 16 AB for a condensing engine, or 15 AB for a non-condensing engine: complete the rectangle, ADEO; then, inasmuch as the area of the rectangle, ABCO, represents the product of the absolute pressure, BC, and volume, AB, of the steam at release, the area of the rectangle, ADEO (= 16 or 15 AB, BC), represents the heat of release, in units of work.

The area, ABHK, of that part of the steam-diagram which lies below the pressure of release, represents a portion of heat saved out of the heat of release, by conversion into mechanical work; and the area, AFGB, of that part of the steam-diagram which lies above the pressure of release, represents an additional expenditure of heat, all of which is converted into work. Hence the following rules:—

* See "A Manual of the Steam-Engine and other Prime Movers;" also papers on the Steam-Engine, in the Philosophical Transactions for 1854.

RULES FOR THE EXPENDITURE AND DISPOSAL OF THE HEAT
(in units of work).

II. Whole heat expended on the steam = area ADEO + area AFG B.

III. Heat converted into mechanical work = area AFG BHK
= area AFG B + ABHK.

IV. Heat rejected with the exhaust steam = area ADEO — area ABHK.

RULE FOR THE EFFICIENCY OF THE STEAM.

V. Efficiency of steam = $\frac{\text{area AFG BHK}}{\text{area ADEO} + \text{area AFG B}}$.

When the volume represented by the length of the line, AB, is equal, or sensibly equal, to the volume swept by the piston, the rule for the efficiency of the steam may be simplified as follows:—

VI. By the methods of Article 33, find the mean absolute pressure of the steam on the piston, the mean back pressure, and their difference, being the mean effective pressure. To the mean absolute pressure add, in condensing engines, 15 times, and in non-condensing engines, 14 times the absolute pressure at release: the sum will be a pressure equivalent to the rate of expenditure of heat; by which last pressure the mean effective pressure is to be divided, to give the efficiency of the steam.

EXAMPLE: Condensing Engine.

	Lbs. on the square inch.
Mean absolute pressure,.....	17.1
Subtract mean back pressure,.....	4.0
Mean effective pressure,.....	13.1
Pressure at release, $6.1 \times 15 =$	91.5
Add mean absolute pressure,.....	17.1
Pressure equivalent to rate of expenditure of heat on the steam,...	108.6
$\frac{13.1}{108.6} = 0.1206$, efficiency of the steam.	

VII. The reciprocal of the efficiency of the steam expresses the proportion in which the mechanical equivalent of the heat expended on the steam must exceed the indicated work of the steam on the piston. In the preceding example, that reciprocal is $\frac{108.6}{13.1} = \frac{1}{0.1206} = 8.29^\circ$

37. Theoretical Steam-diagrams for Proposed Engines.—The curves actually described on the indicator-cards of steam-engines present so many differences as to the mode in which the pressure and volume of the steam vary during its action on the piston, that their figures cannot be expressed exactly by any general system of mathematical formulæ.

In order that it may be possible to compute from theoretical principles the power and efficiency of the fluid in proposed steam-engines, a figure is assumed for the diagram, approximating to the real figure, but more simple (see Fig. 5). In that figure, AB represents the volume of a certain mass of steam when admitted into the cylinder, so as to drive the piston through a space equal to that volume. The first assumption by which the diagram is simplified is, that the pressure of the steam remains constant during its admission, so that AB is a straight line parallel to OX, and the constant absolute pressure is represented by OA = GB.

The curve, BC, represents the expansion of the steam after its admission is cut off. In actual diagrams, that curve presents a great variety of figures, depending upon the communication

of heat to and from the steam, and other causes. The second assumption consists in assigning to the curve, BC, one or other of certain definite figures, according to the following suppositions:—

I. When between an outer casing of slowly conducting materials (such as felt and wood) and the cylinder, there is an inner casing of iron called the "steam-jacket," supplied with steam from the boiler, it is assumed that the heat communicated by means of that jacket to the steam expanding in the cylinder, is just sufficient to prevent any practically appreciable part of it from becoming liquid; so that BC is part of a curve whose co-ordinates represent the pressures and the volumes of a given weight of steam of saturation. The figure of such a curve is shown in Plate $\frac{K}{1}$; and in ordinary cases it approaches nearly to a curve of the hyperbolic class, the pressure varying inversely as the seventeenth power of the sixteenth root of the volume. The same expansion-curve may be obtained without a steam-jacket, if the steam is superheated to an extent sufficient to supply on its admission, to the metal of the cylinder, the heat which the steam-jacket would otherwise supply.

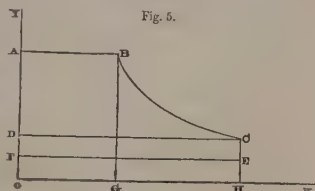
II. With a less perfect action of the steam-jacket, or of superheating, so that a small quantity of steam is first liquefied in the cylinder, and afterwards re-evaporated, the expansion-curve, BC, may be assumed to be sensibly a common hyperbola; the pressure varying inversely as the volume.

The first supposition corresponds to the highest efficiency attainable with steam which is saturated when it works in the cylinder; the second, to the ordinary efficiency of really good engines. If jacketing and superheating are both neglected, the expansion-curve is flattened, and the efficiency diminished still further; but such defects ought not to occur in well-designed engines.

The third assumption is, that the steam is exhausted, or discharged from the cylinder during the return stroke, at a constant pressure; so that the lower side, EF, of the diagram is a straight line parallel to OX; and the constant back pressure is represented by OF = HE, which may be equal to, or less than the pressure at the end of the expansion, HC. (It would be possible, also, to make the back pressure greater than the pressure at the end of the expansion; but this never occurs in engines that are well constructed and well worked.) The third assumption involves also the assumption, that the fall of pressure, if any, at the end of the stroke (represented by CE), takes place suddenly.

The value taken for the assumed constant back pressure ought of course to be equal to the mean value of the actual variable back pressure, so far as it can be accurately ascertained. What that mean value is in different cases will be considered in the next Article.

The fourth assumption consists in neglecting the volume of the liquid water as compared with that of the steam, so that the side, FDA, of the diagram is a straight line coinciding with OY, instead of being a curve having ordinates parallel to OX, representing the successive volumes of the water as it sustains a gradually increasing pressure in the feed-pump. This assumption gives rise to no error appreciable in practice.



* For a detailed explanation of the rules in this Article, and a comparison of their results with those of exact formulæ, see the Engineer for the 5th January, 1866.

Thus is obtained a diagram for purposes of calculation, of the kind of form represented by ABCFEFDA, of which the side, BC, alone is curved.

The effect of compressing or "cushioning" steam will be considered in a later Article.

38. *Estimated Back Pressure.*—If the steam working in steam-engines were unmixed with air, and if it could escape without resistance and in an inappreciably short time from the cylinder after having completed the forward stroke, the back pressure would be simply, in non-condensing engines (conventionally called "*high pressure engines*"), the *atmospheric pressure* for the time; and in condensing engines, the pressure corresponding to the temperature in the condenser. This may be called the *pressure of condensation*.

The mean back pressure, however, always exceeds the pressure of condensation, and sometimes in a considerable proportion. One cause of this, which operates in condensing engines only, is the presence of air mixed with the steam, which causes the *pressure in the condenser*, and consequently the back pressure also, to be greater than the pressure of condensation of the steam. For example, an ordinary temperature in a condenser, when working properly, is about 104° Fahr., to which the corresponding pressure of steam is 1.06 lbs. on the square inch. But the absolute pressure in the best condensers is scarcely ever less than 2 lbs. on the square inch, or nearly *double* of the pressure of condensation.

The principal cause, however, of increased back pressure, is resistance to the escape of the steam from the cylinder, by which, in condensing engines, the mean back pressure is caused to be from 1 to 3 lbs. on the square inch greater than the pressure in the condenser. There is as yet no satisfactory theory of that resistance, so that it cannot be computed for any proposed engine by means of a general formula.

The back pressure, therefore, in proposed condensing engines, can for the present only be estimated roughly from the results of experience in particular cases. The following is a summary of some such results:—

Rate of expansion from 1½ to 8,.....	Mean Back Pressure,	
	Lbs. on the square foot.	Lbs. on the square inch.
" " from 1½ to 3,.....	720	5
" " from 4 to 7,.....	648 to 504	4½ to 3½
" " from 8 to 15,.....	504 to 432	3½ to 3

There is a deficiency of precise experimental data on this subject, because of the frequent omission to observe the atmospheric barometer at the time when the indicator-diagrams of steam-engines are taken. The consequence of that omission is, that the diagrams show only the *effective* pressures of the steam, and not the *absolute* pressures, which are left to be roughly estimated by guessing the probable atmospheric pressure.

39. *Properties of Dry Saturated Steam.*—Rules for calculating the relations between the pressure and volume of dry saturated steam, and between its temperature and its latent and total heat, have been given in Article 23, Subdivision III., and in Article 27, Subdivision III.; and in Article 30 it has been shown how to express the latent and total heat by equivalent quantities of mechanical energy. The most convenient way by far, however, of finding the volume of steam from the pressure, or the pressure from the volume, for practical purposes, is by means of such a diagram as that given in Plate F. Annexed is the table from which that diagram was constructed; and in that table are

contained also the elasticity, or product of the pressure and volume, and the total and sensible heat of steam at different temperatures, all expressed in foot-pounds of work, *the feed-water in each case being assumed to be at the temperature of 104° Fahr. = 40° Cent.* In the eighth column is given the work that one lb. of steam would perform if it were worked expansively in a *jacketed* cylinder down to the pressure corresponding to the assumed temperature of the feed-water.

TABLE OF PROPERTIES OF STEAM.

Temperature.		Absolute Pressure, Lbs. on the square inch.	Volume of One Lb., in cubic feet.	Elasticity of One Lb. Foot-lbs.	Total Heat, from 40° C. = 104° F. Foot-lbs.	Sensible Heat, from 40° C. = 104° F. Foot-lbs.	Total Work of Dry Steam down to 40° C. = 104° F. Foot-lbs.
Cent.	Fahr.						
40	104	1.06	312.8	47720	804181	0	0
50	122	1.78	192.0	49159	808412	13910	25060
60	140	2.88	122.0	50559	812642	27847	49050
70	158	4.51	80.02	51940	816872	41799	72050
80	176	6.86	53.92	53248	821103	55751	94120
90	194	10.16	37.26	54513	825333	69746	115310
100	212	14.70	26.36	55783	829563	83751	135660
110	230	20.80	19.03	56964	833793	97800	155190
120	248	28.83	14.00	58130	838023	111848	174000
130	266	39.25	10.48	59238	842254	125952	192080
140	284	52.52	7.973	60298	846484	140009	209490
150	302	69.21	6.153	61320	850715	154273	226230
160	320	89.86	4.816	62330	854945	168475	242380
170	338	115.1	3.814	63256	859175	182746	257930
180	356	145.8	3.057	64180	863405	197046	272910
190	374	182.4	2.476	65043	867635	211401	287360
200	392	225.9	2.025	65857	871866	225782	301290
210	410	276.9	1.672	66665	876096	240262	314736
220	428	336.3	1.393	67468	880327	254769	327690
(1.)	(2.)	(3.)	(4.)	(5.)	(6.)	(7.)	(8.)

To find the whole expenditure of heat on a lb. of steam under the supposed circumstances, add the whole work to the total heat (in foot-lbs.) at the temperature of the feed-water.

The theoretical principles according to which the numbers in the eighth column of the preceding table are calculated, are explained in a paper in the Philosophical Transactions for 1854; and also in "A Manual of the Steam-engine and other Prime Movers."

40. *Estimation of Pressures in Proposed Engines.*—RULE I. A.

—To calculate the ratio of the mean absolute pressure to the absolute pressure of admission, when the steam is supposed to expand in the dry saturated condition, agreeably to Supposition I. of Article 37—From 17 subtract 16 times the reciprocal of the sixteenth root^c of the rate of expansion, and divide the remainder by the rate of expansion: the quotient will be the ratio of the mean absolute pressure to the pressure of admission.

EXAMPLE.—Rate of expansion, 5.

$$\sqrt[5]{5} = 2.2361; \sqrt[5]{2.2361} = 1.4953; \sqrt[5]{1.4953} = 1.2228;$$

$$\sqrt[5]{1.2228} = 1.1058.$$

$$\text{Reciprocal of sixteenth root, } \frac{1}{1.1058} = 0.9043$$

$$\times 16$$

$$14.4688$$

$$\text{The above to be subtracted from } 17.0000$$

$$\text{Divide by rate of expansion, } 5) 2.5312 \text{ remainder.}$$

$$\text{Required ratio of mean absolute pressure to pressure of admission, } 0.5062$$

RULE I. B.—To solve the same question by the aid of the diagram in the upper right-hand corner of Plate F.—Take the reciprocal of the rate of expansion, or *effective cut-off*; if it is

^c That is, the square root of the square root of the square root of the square root. This calculation is easily made by the aid of a table of squares.

less than 0.5, look for it at the lower edge of the diagram; if greater, at the upper edge. The height of the curve above the lower edge of the diagram at the point so found will show the required ratio. Each horizontal division represents $\frac{1}{16}$ of the volume swept by the piston; each vertical division, $\frac{1}{16}$ of the absolute pressure of admission.

RULE I. c.—On the same supposition, to find the probable ratio of the absolute pressure of release to the absolute pressure of admission—Divide the reciprocal of the sixteenth root of the rate of expansion by the rate of expansion.

EXAMPLE.—Rate of expansion, 5.
Reciprocal of sixteenth root, as before, 0.9043
 $\frac{0.9043}{5} = 0.1809$, ratio required.

RULE II. a.—To calculate the ratio of the mean absolute pressure to the absolute pressure of admission, when the expansion-curve is supposed to be a common hyperbola, agreeably to Supposition II. of Article 38—Add 1 to the hyperbolic logarithm^c of the rate of expansion, and divide by the rate of expansion; the quotient will be the required ratio.

EXAMPLE.—Rate of expansion, 5.
Hyperbolic logarithm of 5, 1.609
Add, 1.000
Divide by rate of expansion, 5) 2.609
Required ratio, 0.522

RULE II. b.—To make the same calculation in the absence of tables—Take 1 to represent the absolute pressure of admission, and compute a series of absolute pressures on the supposition that after the cut-off they vary inversely as the volumes; then find the mean absolute pressure (in fractions of the absolute pressure of admission) by Simpson's First Rule. (See Division I., Article 20, page 13.)

EXAMPLE.—Rate of expansion, 5; number of intervals, 10.

Divisions of Stroke.	Pressures.	Simpson's Multipliers.	Products.
0	1.000	1	1.000
1	1.000	4	4.000
Cut-off, 2	1.000	2	2.000
3	0.667	4	2.667
4	0.500	2	1.000
5	0.400	4	1.600
6	0.333	2	0.667
7	0.286	4	1.143
8	0.250	2	0.500
9	0.222	4	0.889
10	0.200	1	0.200

Divide by $10 \times 3 = 30$) 15.666

Required ratio, 0.522

RULE II. c.—The ratio of the absolute pressure of release to that of admission, according to Supposition II., is equal to the effective cut-off, or reciprocal of the rate of expansion.

EXAMPLE.—Rate of expansion, 5.
 $\frac{\text{Absolute pressure of release}}{\text{Absolute pressure of admission}} = \frac{1}{5} = 0.2$

Suppositions I. and II. differ but little in practice; and the choice between them in making calculations for a proposed engine is in most cases a matter of convenience: Supposition I. being best when the computer has the aid either of a table of squares, or of the diagrams in Plate $\frac{F}{7}$; and Supposition II. the more convenient either in the absence of diagrams and tables, or when a table of hyperbolic logarithms only is at hand.

^c If tables of common logarithms alone are at hand, multiply the common logarithm by 2.3026; the product will be the hyperbolic logarithm.

The rules of Supposition II. may be used as rough approximations, when steam is worked expansively in a cylinder without jacketing or superheating, provided the rate of expansion does not exceed 2; but when the rate of expansion in such cases exceeds 2, no trustworthy rules can be laid down.

The following table gives examples of the results of calculations of the ratio of the mean absolute pressure to the absolute pressure of admission, by Rules I. a and II. a respectively:—

Rate of Expansion.	Effective Cut-off.	Mean Pressure \div Pressure of Admission, according to	Supposition I.	Supposition II.
20	0.5	186	200
13 $\frac{1}{2}$	0.75	254	269
10	1	314	330
8	1.25	370	385
6 $\frac{2}{3}$	1.5	417	435
5	2	506	522
4	2.5	582	596
3 $\frac{1}{2}$	3	648	661
2 $\frac{2}{3}$	3.5	707	717
2 $\frac{1}{2}$	4	756	765
2 $\frac{1}{3}$	4.5	800	809
2	5	840	846
1 $\frac{5}{7}$	5.5	874	878
1 $\frac{4}{5}$	6	900	906
1 $\frac{3}{5}$	6.5	926	929
1 $\frac{2}{5}$	7	945	950
1 $\frac{1}{5}$	7.5	960	965
1 $\frac{1}{4}$	8	976	978
1 $\frac{1}{3}$	8.5	986	989
1 $\frac{1}{2}$	9	997	997
1	10	1000	1000

41. *Estimation of Efficiency of Steam in Proposed Engines.*—To estimate the probable efficiency of the steam in a proposed engine, proceed as follows:—

STEP I.—Estimate the mean absolute pressure by the rules of Article 40 of this Division, and the mean back pressure by the principles of Article 38: the difference between them will be the *mean effective pressure*.

STEP II.—Estimate the absolute pressure of release by the rules of Article 40; multiply it, if the engine is condensing, by 15, or if non-condensing, by 14; to the product add the mean absolute pressure: the sum will be a *pressure equivalent to the rate of expenditure of heat on the steam*.

STEP III.—Divide the mean effective pressure by the pressure just mentioned: the quotient will be the probable efficiency of the steam.

EXAMPLES.—Suppose absolute pressure of admission, 40 lbs. on the square inch; mean back pressure, 4 lbs. on the square inch; rate of expansion, 5.

	Supposition I.	Supposition II.
STEP I.—Absolute pressure of admission,.....	40	40
× Ratio (by Rule I. a or II. a of Article 40),.....	5.06	5.22
Mean absolute pressure,.....	20.24	20.88
Subtract mean back pressure,.....	4.00	4.00
Mean effective pressure,.....	16.24	16.88
STEP II.—Mean absolute pressure,.....	40	40
× Ratio (by Rule I. c or II. c of Article 40),.....	0.1809	0.2
Absolute pressure of release,.....	7.236	8.000
× Multiplier for a condensing engine,.....	15	15
Add mean absolute pressure,.....	108.54	120.00
Pressure equivalent to rate of expenditure of heat,.....	20.24	20.88
STEP III.—.....	Divide 16.24	16.88
Quotient, being the probable efficiency of the steam,.....	by 128.78	140.88
	0.126	0.120

42. *Excess of Pressure in Boiler above Pressure of Admission.*

—The fall which the pressure of the steam undergoes during its passage from the boiler to the cylinder, is due to the following causes:—

I. The resistance of the *steam-pipe*, through which the steam passes from the boiler to the valve-box.

II. The resistance of the *regulator*, or *throttle-valve*, by which the steam-pipe is sometimes partially closed.

III. The resistance of the *ports*, or steam-passages through which the steam is admitted from the valve-box into the cylinder, and which are at times partially closed by the valves, so as to have their resistance increased.

In a well-constructed engine the steam-pipe should be so proportioned, that supposing the density of the steam to be the same in it and in the cylinder, the velocity of the steam through the steam-pipe shall not exceed about 100 feet per second; and then the resistance in the pipe may be neglected.

The resistance of the regulator in a properly proportioned steam-pipe is inappreciable when it is wide open; and when it is partially closed, the investigation of mathematical relations between the resistance and the opening is practically unimportant, because the extent of opening of the regulator required to produce any given reduction of pressure in any existing engine can easily be found by trial.

There remains to be considered, the resistance of the cylinder-ports. In all ordinary examples of marine engines, the effect of that cause may be estimated with sufficient accuracy for practical purposes by the following

RULE.—Multiply the square of the mean velocity of the piston by the square of the ratio in which the area of the piston exceeds the area of the steam-port; divide the product by 180 times the absolute temperature of the steam in degrees of Fahrenheit (or by 324 times the absolute temperature in Centigrade degrees): the quotient will be the fraction of the absolute pressure in the boiler which is lost by the steam in passing into the cylinder.*

EXAMPLE.—Suppose absolute pressure in boiler, 39.25 lbs. on the square inch; absolute temperature, $266^{\circ} + 461^{\circ}.2 = 727^{\circ}.2$ Fahr.; area of piston \div area of port = 25.

$$\text{Then } \frac{4^2 \times 25^2}{180 \times 727.2} = \frac{1}{13} \text{ nearly;}$$

that is to say, the steam loses one-thirteenth of its absolute pressure in passing from the boiler to the cylinder; and consequently,

$$\text{Pressure of admission} = 39.25 \times \frac{12}{13} = 36.23 \text{ lbs. on the square inch.}$$

It appears further, from the experiments of Mr. Clark, that the loss of pressure of misty steam in traversing passages exceeds that of dry steam in a proportion which cannot be computed with any approach to precision, but which ranges from $1\frac{1}{4}$ to $2\frac{1}{2}$, and sometimes even to 3.

The fall of pressure which occurs during the passage of steam from the boiler to the cylinder, does not wholly represent *wasted energy*; for being expended in friction, it produces heat; so that steam which has had its pressure lowered by the resistance of passages, or as it is called, has been *wire-drawn*, is superheated. (In the example just given, the steam would be superheated about 4° or 5° Fahr.) Even supposing, however, that no energy is directly wasted when steam is wire-drawn, there is still an

indirect waste of energy from the lowering of its pressure, which, by diminishing the absolute pressure upon the piston as compared with the back pressure, and by diminishing the extent of expansive working of which the steam is capable, lowers its efficiency.

43. *Effects of Disturbing Causes on Diagrams.*—The deviations of the steam-diagram from the ideal form described in Article 37, which are now to be considered, are not those produced by oscillation or friction in the indicator, but those really due to the action of the steam. They are as follows:—

I. *Wire-drawing at Cut-off.*—The valve by which the steam is admitted into one end of the cylinder, closes, in order to cut off the admission of steam, not instantaneously, but by degrees, especially when it is a slide-valve. In consequence of this, the loss of pressure by the steam in passing from the valve-chest into the cylinder gradually increases, and the pressure of the steam in the cylinder begins gradually to diminish, before the complete closing of the valve; so that the top of the diagram, which is drawn during the admission of the steam, instead of presenting a straight line, A B (Fig. 6), parallel to O X, presents a drooping curve, convex upwards, such as A H G.

The point of the stroke where the *complete closing* of the valve, or *actual cut-off*, takes place, is usually marked on the diagram by a *point of contrary flexure*, G, where the curve convex upwards, H G, produced by wire-drawing, touches the curve of expansion, G C, which is concave upwards. The steam begins to a certain extent to work expansively before the valve is completely closed, and the energy exerted is nearly the same as if the valve closed instantaneously at a somewhat earlier point of the stroke, which may be called the *virtual*, or *effective cut-off*. To find approximately that point, produce the expansion-curve, C G, upwards, and draw the straight line, A B, to meet it; then the point, B, marks the effective cut-off, and determines the effective ratio of expansion to be used in computing the efficiency.

II. *Clearance* is a term used to include, not merely the clearance proper, which is the space between the piston and the end of the cylinder to which it is nearest at the end or beginning of a stroke, but also the volume of the ports, and generally the whole *minimum* space between the piston and the valves. That space, as well as the space through which the piston sweeps, has to be filled with steam.

The clearance, for purposes of calculation, is expressed in the form of a fraction of the space swept through by the piston during a single stroke. That fraction ranges from $\frac{1}{8}$ to $\frac{1}{40}$, and sometimes less, in different engines, being greatest in the smallest engines. The equivalent length of cylinder varies less, being usually from one to two inches.

The clearance affects the ratio of expansion in the following manner:—

RULE A.—To the fraction of the stroke at which the steam is cut off, add the fraction expressing the clearance; divide the sum by 1 + the latter fraction: the quotient will be the *effective cut-off*, and its reciprocal, the *real rate of expansion*.

* The constant divisor is deduced from Mr. D. K. Clark's experiments.

EXAMPLE.—Suppose actual cut-off, 0.16; clearance, 0.05.

$$\text{Then effective cut-off} = \frac{0.21}{1.05} = 0.2;$$

$$\text{Real rate of expansion} = \frac{1.05}{0.21} = 5.$$

When no arrangement is made for preventing the additional expenditure of steam required for filling the clearance, the mean effective pressure and the efficiency of the steam are diminished according to the following rules:—

RULE B.—From the mean effective pressure, as calculated from the true rate of expansion by the rules of Article 40, subtract the product of the clearance into the difference between the pressure of admission and the mean pressure: the remainder will be the mean effective pressure corrected for clearance.

RULE C.—The pressure equivalent to the rate of expenditure of heat on the steam (computed as in Article 41) is increased nearly in the ratio of 1 + the clearance : 1.

EXAMPLE.—Suppose clearance, 0.05 of stroke; absolute pressure of admission, 40 lbs. on the square inch; mean back pressure, 4 lbs. on the square inch; real rate of expansion, 5.

Mean effective pressure, by Rule I. A of Article 40,	16.24
Subtract $(40 - 16.24) \times 0.05 =$	1.19
Corrected mean effective pressure,	15.05
Pressure equivalent to rate of expenditure of heat, computed as in Article 41,	128.78
..... $\times 1.05$	
The same, corrected for clearance,	135.22
Efficiency of steam, corrected for clearance,	$\frac{15.05}{135.22} = 0.111;$
(instead of 0.126.)	

III. *Compression, or cushioning*, is effected by closing the eduction-valve before the end of the return stroke; for example, at the point corresponding to M, Fig. 6. This confines a certain quantity of steam in the cylinder, which is compressed by the piston during the remainder of the return stroke, the rise of its pressure being represented by some such curve as M A. In the figure, that curve is made to terminate at A, in order to represent the most advantageous adjustment of the compression, which takes place when the quantity of steam confined or "cushioned" is just sufficient to fill the clearance at the initial pressure. That adjustment saves the steam which would otherwise be wasted in filling the clearance, and thus prevents the efficiency of the steam from being diminished by clearance; the mean effective pressure, and the expenditure of heat per stroke, being diminished in the same proportion: but it can seldom be effected in marine engines, because of the lowness of the back pressure. When it is effected, the ratio in which the mean effective pressure, and the expenditure of heat per stroke, are both diminished, may be roughly approximated to as follows:—Multiply the fraction expressing the clearance by the *apparent* rate of expansion (or reciprocal of the actual cut-off); to the product add 1, and take the reciprocal of the sum.

IV. *Release* (already referred to in Article 36) means opening the exhaust-port for the escape of the steam before the forward stroke is finished, in order to diminish the back pressure. In an engine in which there is no release (the exhaust-port opening exactly at the end of the forward stroke), the diagram during the return stroke is usually a curve more or less resembling the dotted line, C M K; the lower side of the ideal diagram used in calculation being a straight line, E F, so placed that its constant ordinate is equal to the mean ordinate of the curve. L K I is

a straight line, whose ordinate, O L, represents the pressure in the condenser (or in non-condensing engines, the atmospheric pressure). By making the release occur early enough, for example, at the point corresponding to P in the diagram, the entire fall of pressure may be made to take place towards the end of the forward stroke, so as to make the back pressure coincide sensibly with that corresponding to the ordinate of K I; and then the end of the diagram will assume a figure represented by the dotted line, P I, which is usually more or less concave upwards. Energy will be saved to the amount represented by the rectangle, $\overline{K F} \times \overline{K I}$, and energy lost to the amount represented by the area of the figure, P C I P; and on the whole, energy will be saved or lost according as the former or the latter of those areas is the larger. The greatest saving of energy is insured by making the release take place at a point, Q, such, that about one-half of the fall of pressure shall take place at the end of the forward stroke, and the other half at the commencement of the return stroke, as indicated by the dotted curve, Q R S.

V. *Conduction of heat to and from the metal of the cylinder*, or

VI. *To and from liquid water contained in the cylinder*, has the effect of lowering the pressure at the beginning, and raising it at the end of the stroke, in the manner already mentioned in Article 36. The general nature of the change thus produced in the diagram is shown by the dotted line, G H I C F, in Fig. 7. Its bad effect in wasting heat, and the remedy for that evil by jacketing or by superheating, have already been mentioned in Article 36.

44. *Cylinder-capacity of Proposed Engines*.—To determine the total cylinder-capacity required for the proposed engine of a given vessel, proceed as follows:—

I. Estimate the probable indicated engine-power required at the intended greatest speed, by the rules of the First Division, Article 164, page 84; having regard to the probable efficiency of the propeller, as determined by the rules of the First Section of the First Chapter of this Division, Articles 6 to 10, pages 248 to 251; and also, if necessary, to the notes at pages 199 and 248.

II. Estimate, by the rules of the Section just referred to, the probable greatest speed of the propeller corresponding to the intended greatest speed of the vessel.

III. Divide the probable greatest speed of the propeller per minute by the advance of the propeller per revolution; that is, by the pitch of a screw, by the circumference of a feathering paddle-wheel measured through the paddle-journals, or by the circumference of a radial paddle-wheel measured round the outer edges of the paddles: the quotient will be the probable greatest number of revolutions of the propelling instrument per minute.

In a paddle-wheel engine, or a screw-engine without gearing, this will also be the number of revolutions of the engine per minute; to find the latter number in a geared screw-engine, divide the number of revolutions of the screw per minute by the *multiple of gearing*; that is, the ratio of the number of teeth in the wheel to the number of teeth in the pinion.

IV. Having reduced the probable indicated engine-power to foot-pounds per minute, divide by the probable number of revolutions of the engine per minute: the quotient will be the probable *indicated work per revolution*, in foot-pounds.

Fig. 7.



V. Divide the indicated work per revolution by *twice* the intended mean effective pressure (determined by the rules of Article 40: the quotient will be the required *cylinder-capacity*. If the mean effective pressure is expressed in lbs. on the square foot, that capacity will be in cubic feet; if the pressure is in lbs. on the square inch, the capacity will be in 144ths of a cubic foot: that is, prisms one foot long and one inch square.

By the capacity of a cylinder, in this rule, is to be understood the mean between the volumes swept through by the piston during a forward stroke and a return stroke respectively.

If an engine has two or more cylinders receiving steam directly from the boiler, and discharging it into the condenser or the atmosphere, the cylinder-capacity found by the rule is the *sum* of the capacities of those cylinders.

But when, in a double-cylindere engine, the steam begins its work in a small cylinder and completes it in a large cylinder, the *large cylinder alone* is included in the capacity found by the rule; and by the mean effective pressure is to be understood, what that pressure would be if the steam worked in the large cylinder alone.

VI. In a double-cylinder engine, to find what proportion the capacity of the small cylinder ought to bear to that of the large cylinder, in order that half the indicated work of the steam, or nearly so, may be done in each of those cylinders—Divide the cube root of the rate of expansion by the rate of expansion. This is but a roughly approximate method; but it is sufficient for practical purposes in ordinary cases.

EXAMPLE.—(From an existing paddle-steamer.)

I. Augmented surface, 8600 square feet; intended greatest speed, 12 knots; estimated indicated horse-power, $8600 \times 12^3 \div 20,000 = 743$. Estimated power, in foot-pounds per minute, $743 \times 33,000 = 24,519,000$.

II. Speed of paddles, $12 \times 1.28 = 15.36$ knots = 1556 feet per minute.

III. The above speed divided by circumference round paddle-journals, 64.4 feet, gives for the number of revolutions per minute, 24.16.

IV. Estimated power, 24,519,000 foot-pounds per minute \div revolutions, 24.16 per minute = 1,014,860 foot-pounds work per revolution.

V. Estimated mean effective pressure, 13 lbs. on the square inch; $1,014,860 \div (2 \times 13) = 39,033$, required total capacity of large cylinders in prisms of one foot long and one inch square. (The actual capacity is 39,016, showing a difference immaterial in practice.)

VI. The engines being double-cylindere, and the mean rate of expansion 5, the capacity of the small cylinders ought to be $\sqrt[3]{5} \div 5 = 0.35$, nearly, of that of the large cylinders, in order to cause one-half of the work of the steam, or nearly so, to be done in the small cylinders. The small cylinders are 0.4 of the capacity of the large cylinders; and about 0.55 of the work is done in them, and 0.45 in the large cylinders.

45. *Estimation of Feed-water and Condensation-water.*—The *net* feed-water required by the boiler of an engine is equal in weight to the steam usefully expended. The *total* feed-water supplied includes a large allowance for leakage, priming, blowing-off, liquefaction in the cylinder, and other losses; and when salt-water is used, for the discharge of brine from the boiler.

The *net* condensation-water is the exact quantity required to

abstract the heat which the steam gives out in the act of condensation. The *total* condensation-water includes an allowance for losses and contingencies.

RULE I.—To calculate the net weight, in lbs., of feed-water required per stroke—Divide the volume of steam expended per stroke at the instant of release, by the volume of one lb. of steam at the pressure of release, as found by the rules of Article 23, Subdivision III., or by the use of the diagram, Plate $\frac{K}{7}$.

RULE I.A.—For a rough approximation, correct to about 10 per cent., and erring on the side of excess only, calculate the product of the pressure and volume of release in foot-pounds (as in Article 36), and divide by 50,000. For the approximate *net volume in cubic feet per stroke*, divide the same product by 3,000,000.

RULE II.—For the gross quantity of feed-water required by a boiler supplied with *pure water*, multiply the net feed-water by 2.

If the supply is *ordinary fresh water*, multiply the net feed-water by $2\frac{1}{2}$.

If the supply is *sea-water*, and the brine is to be discharged from the boiler at *n* times the saltiness of sea-water, multiply the net feed-water by $\frac{2n}{n-1}$; for example, suppose

$$\begin{array}{lcl} \frac{\text{Saltiness of brine discharged}}{\text{Saltiness of sea-water}} = 3 & \dots\dots & 2\frac{1}{2} \dots\dots 2. \\ \text{Then } \frac{\text{gross}}{\text{net}} \text{ feed-water} = 3 & \dots\dots & 3\frac{1}{2} \dots\dots 4. \end{array}$$

RULE III.—To calculate the *net* weight, in lbs., of condensation-water required per stroke—Divide the mechanical equivalent of the rejected heat per stroke (see Article 36 and Article 41) by the difference between the temperature of condensation and the temperature at which the cold water is obtained, and by Joule's equivalent (see Article 30).

Assuming the difference of temperature to be 45° Fahr. = 25° Cent., the divisor is found to be 35,000, nearly, for British measures, or 10,600 for French measures.

For *cubic feet per stroke*, use 2,200,000 as the divisor.

RULE IV.—For the gross supply of condensation-water, multiply the net quantity by 2.

45A. *Adjustment of Pressures in Vertical or Inclined Cylinders.*—In a vertical, or nearly vertical, cylinder of a direct-acting engine (such as that shown in Plate $\frac{N}{7}$), it is favourable to steadiness of motion that only half the weight of the connecting-rod should be balanced along with the weight of the crank, by means of a counterpoise fixed to the shaft; leaving the remaining half of the weight of the connecting-rod, together with the whole weight of the piston-rod and piston, to form a force, of which the whole, when the cylinder is vertical, or one of its components, when the cylinder is inclined, acts alternately with the steam during the down-stroke, and against the steam during the up-stroke. To prevent unsteadiness of motion from the action of this force, the mean absolute pressure on the bottom of the piston should be made greater, and the mean absolute pressure on the top of the piston less, than the general mean absolute pressure, according to the following

RULE.—Find the probable weights of the piston and piston-rod, and half the connecting-rod; multiply their sum by the cosine of the angle at which the cylinder leans from the vertical (if the cylinder is vertical, that cosine is 1); divide the force thus found by the area of the piston; add the quotient to, and

subtract it from, the general mean absolute pressure, for the mean absolute bottom pressure and top pressure, respectively.

Divide those pressures by the absolute pressure of admission, and find the values of the effective cut-off corresponding to the ratios so calculated, by means of the smaller diagram in Plate $\frac{K}{T}$.

EXAMPLE.—Given, the probable weight of piston, piston-rod, and half connecting-rod, 14,696 lbs.

Cylinder leans 45° from vertical; $\cos. 45^\circ = 0.7071$; $14,696 \times 0.7071 = 10,392$ lbs.

Given also, area of piston, 4600 square inches;

Then $\frac{10392}{4600} = 2.26$ lbs. on the square inch.

Let general mean absolute pressure = 17.2 lbs. on the square inch; and let absolute pressure of admission = 34 lbs. on the square inch:

Then $17.2 + 2.26 = 19.46$, mean absolute bottom pressure;

$17.2 - 2.26 = 14.94$, mean absolute top pressure;

$19.46 \div 34 = 0.57$, corresponding to 0.24, effective cut-off at bottom;

$14.94 \div 34 = 0.44$, corresponding to 0.16, effective cut-off at top.

SECTION II.—CONSTRUCTION OF MARINE ENGINES.

46. *General Reference to Plates.*—The Plates $\frac{N}{T}$, $\frac{N}{S}$, representing the engines of the *Arabia* (made by Messrs. R. Napier & Sons), are chosen to illustrate in detail the construction of marine engines; because, being of the class called "side-lever" engines, they contain all the essential parts that are found in direct-acting engines, together with some parts that are peculiar to the side-lever arrangement.

Plate $\frac{N}{T}$ shows a longitudinal section of an engine. Plate $\frac{N}{S}$ shows three transverse sections, through the air-pump, condenser, and valve-casing, respectively.

The general arrangement of the engines and boilers of the *Arabia* is the same as in the *Persia*, and is shown in Plates $\frac{A}{S}$ and $\frac{A}{S}$. The principal dimensions of the engines of those two ships will be given in a table in the course of this Article.

REFERENCES TO THE ENGINES, PLATES $\frac{N}{T}$, $\frac{N}{S}$.

A Foundation or sole-plate.	F ⁷ Air-pump rod.
B Cylinder.	F ⁸ " cross-head.
B ¹ " cover and stuffing-box.	F ⁹ " guide-rods.
B ² " upper port or nozzle.	F ¹⁰ " side-rods.
B ³ " lower port or nozzle.	G Hot-well.
B ⁴ " piston.	G ¹ " air-chamber.
B ⁵ " piston packing-ring and springs.	G ² " discharge-pipe.
B ⁶ " piston junk-ring.	G ³ " air-pipe.
B ⁷ " piston-rod.	H Feed-pump.
B ⁸ " cross-head.	H ¹ " plunger.
B ⁹ " side-rod journals.	H ² " valve-chest.
C Slide-valve or steam-valve.	J Bilge-pump.
C ¹ " casing.	J ¹ " plunger.
C ² " casing-cover and stuffing-box.	K Main-centre.
C ³ " casing faucet-joint.	K ¹ " casing.
C ⁴ " packing and pinching-pins.	L Side-levers.
C ⁵ Steam expansion-valve-chest.	L ¹ " keys for tightening bearings.
C ⁶ " pipe.	L ² " studs for cylinder-side-rods.
C ⁷ " throttle-valve-shaft.	L ³ " studs for cross-tail-links.
D Condenser.	L ⁴ " studs for air, feed, and bilge pumps.
D ¹ " passage from upper-port.	L ⁵ " stud for additional bilge pump.
D ² " passage from lower-port.	L ⁶ Side-lever stud for deck-pump.
D ³ " passage to air-pump.	L ⁷ " stud for water-closet-pump.
E Injection-rose.	L ⁸ " stud for parallel-motion.
E ¹ " sluice.	M Connecting-rod.
E ² " sluice-shaft.	M ¹ " cross-tail.
F Air-pump.	M ² " cross-tail-links.
F ¹ " cover and stuffing-box.	N Crank.
F ² " bucket.	N ¹ " pin.
F ³ " bucket-valve.	N ² " shaft.
F ⁴ " delivery-valve.	O Plumber-block entablature.
F ⁵ " delivery-valve-guides.	
F ⁶ " foot-valve.	

O ¹ Plumber-block cover.	S ⁹ Valve-shaft stops.
O ² " bolts and nuts.	T Expansion-valve.
O ³ " oil-cup.	T ¹ " valve-shaft with levers.
P " columns.	T ² " valve disconnecting handle.
P ¹ " column-sockets.	T ³ " cam on crank-shaft.
P ² " column-stays.	T ⁴ " cam-tumbler, or pulley-frame with counterbalance.
P ³ " column-cross-stays.	T ⁵ " connecting-rod.
Q Diagonal-framing.	U Starting-wheel-shaft.
Q ¹ " stay.	U ¹ " bearings.
Q ² " stay.	V Blow-through-valve-chest.
R Parallel-motion-shaft.	W Snifting-valve.
R ¹ " radius-levers.	W ¹ " discharge grating.
R ² " radius-rods.	W ² " pipe.
R ³ " connecting-rod.	X Priming or escape-valves with springs, for spaces above and below piston.
R ⁴ " shaft-bushes.	Y Keelsons for supporting sole-plate and boilers, extending the whole length of the engine-room.
S Valve-shaft.	Z Counter on one engine, clock on the other.
S ¹ " lever or lifter.	
S ² " clutch.	
S ³ " links.	
S ⁴ " spindle.	
S ⁵ " guides.	
S ⁶ " counterbalance.	
S ⁷ " counterbalance-lever.	
S ⁸ " connection-rods.	

PRINCIPAL DIMENSIONS OF ENGINES.

VESSELS.	Arabia.	Persia.
1 Builder of vessel,	R. Steele,	R. Napier & Sons.
2 Material of vessel,	Wood,	Iron.
3 When completed,	1852,	1855.
4 Length, keel and fore-rake,	285 ft. 0 in.	360 ft. 0 in.
5 Breadth of beam,	40 ft. 8 in.	45 ft. 0 in.
6 Breadth over paddle-boxes,	66 ft. 4 in.	71 ft. 4 in.
7 Depth of hold,	27 ft. 6 in.	29 ft. 8 in.
8 Depth over planking,	30 ft. 10 in.	32 ft. 8 in.
9 Tonnage, builder's measurement,	2292 $\frac{3}{4}$ tons,	3586 $\frac{3}{4}$ tons.
10 Tonnage, new measurement,	2402 $\frac{9}{10}$ "	3300 $\frac{3}{10}$ "
11 Tonnage of engine-room, do.,	928 $\frac{3}{10}$ "	1221 $\frac{1}{10}$ "
12 Tonnage, register, do.,	1474 $\frac{6}{10}$ "	2079 $\frac{2}{10}$ "
13 Length on deck, do.,	284.4 feet,	366.6 feet.
14 Breadth on deck, do.,	37.4 "	45.0 "
15 Depth of hold, do.,	27.7 "	29.9 "
16 Length allowed for engine-space,	82.8 "	115.6 "
17 Draught, mean (one-half of coals consumed),	19 ft. 0 in.	21 ft. 6 in.
18 Area of midship-section at mean draught,	686 sq. ft.	810 sq. ft.
19 Displacement at mean draught,	3950 tons,	5250 tons.
ENGINES.		
20 Kind of engines,	Lever,	Lever.
21 Material of engine-framing,	Malleable-iron,	Malleable-iron.
22 Collective nominal horse-power, } per Admiralty Rule,	873	850
23 Cylinders, diameter and stroke,	103 in. 9 ft. 0 in.	100 $\frac{1}{2}$ in. 10 ft. 0 in.
24 Air-pump, diameter and stroke,	60 $\frac{1}{2}$ in. 4 ft. 7 in.	60 $\frac{1}{2}$ in. 5 ft. 1 in.
25 Length of sole-plate, extreme,	80 ft. 0 in.	30 ft. 3 in.
26 Length of side-levers, between centres,	21 ft. 11 in.	22 ft. 1 in.
27 Distance between centres of cylinders,	15 ft. 10 in.	15 ft. 10 in.
28 Distance between centres of levers,	11 ft. 3 in.	11 ft. 3 in.
29 Height of main-centre,	3 ft. 8 in.	4 ft. 5 in.
30 Height of paddle-shaft,	22 ft. 9 in.	24 ft. 3 in.
WHEELS.		
31 Paddle-wheels, diameter over floats } (all radial),	35 ft. 6 in.	38 ft. 6 in.
32 Number and dimensions of floats } (one wheel),	28, 10 ft. 6 in. 3 2	28, 10 ft. 8 in. 2 0

The furnaces and boilers will be described in Chapter III. of this Division.

The materials of the parts of those engines are as follows:—

CAST-IRON.—Sole-plates, condensers, cylinders, side-levers (in the *Arabia*), diagonal-frames, entablatures, plumber-blocks, hot-wells, air-chambers, covers, slide-valves, valve-casings, pumps, flooring-plates, starting-platform, pipes of bilge-pumps and of water-closet pumps.

WROUGHT-IRON.—Columns, stays, intermediate and paddle-shafts, side-levers (in the *Persia*), all parts of the mechanism not otherwise specified.

BRASS (or more properly BRONZE).—Pistons (packing and junk-rings of cast-iron); air-pump chambers, buckets, and valves; pump-plungers, port-faces, valve-guides; expansion, injection, blow-through, and escape valves; journal bushes.

COPPER.—Pipes, except those of bilge and water-closet pumps.

STEEL.—Springs for piston-packing, and for escape and surplus valves.

In the sections, the darkest tint represents cast-iron; the medium, wrought-iron; the lightest, brass.

Plates $\frac{1}{2}$ and $\frac{3}{4}$ also are lettered, but on the principal parts of the engines only; the object in those Plates being to illustrate general arrangement rather than details.

DESCRIPTION OF HORIZONTAL SCREW-ENGINES IN PLATE $\frac{1}{2}$.—The Plate represents a set of three expansive horizontal screw-propeller engines, by Messrs. Maudslay, Sons, & Field. Collective nominal horse-power, 150. The boilers, as well as the engines, are shown; but the boilers will be described in Chapter III.

H, H, Steam-pipe.

K, Throttle-valve.

L, L, L, Slide-valve-chests of the three cylinders.

M, M, M, The three cylinders, whose three pistons drive three cranks, making angles with each other of one-third of a revolution. Each cylinder has two piston-rods, one passing above, and the other below the shaft. Beyond the shaft, the two piston-rods are connected together by an oblique cross-head, sliding in guides; and from that cross-head a connecting-rod comes back to the crank.

N, N, N, Exhaust-pipes, leading from the three slide-valve-chests to

O, O, Two surface-condensers; each condensing the steam from one of the endmost cylinders, and half the steam from the middle cylinder.

P, Cold water supply-pipe.

Q, Q, Cold water discharge-pipes.

R, Cold water suction-pipe.

S, S, Condensed-water pipes, leading to

T, Hot-well, or feed-water tank.

U, Main plumber-block of shaft.

V, Worm-wheel, for turning shaft by hand. (This wheel gears with an endless screw.)

W, Toothed-wheel, for driving the eccentric-shaft. (The eccentric-shaft is shown in the thwartship section at engines only.)

X, Reversing-gear. (Shown in longitudinal section only.)

Y, One of the slide-valve-rods. (Shown in thwartship section only. There are a pair of those rods to each slide-valve.)

Z, Z, Air-pumps.

A a, Cold water or "circulating" pump.

DESCRIPTION OF VERTICAL INVERTED DOUBLE-CYLINDER GEARED SCREW ENGINES, IN PLATE $\frac{3}{4}$.—The Plate represents a pair of vertical inverted double-cylinder expansive geared screw-propeller engines, by Messrs. Randolph, Elder, & Co. Collective nominal horse-power, 250. Indicated horse-power, variable at pleasure from 500 to 1000. The view in the Plate is a vertical thwartship section through the two cylinders of the forward engine. The after engine is exactly similar. The engines drive two cranks, which are at right angles to each other.

A, High-pressure cylinder, } Both jacketed at sides, top, and bottom.
B, Low-pressure cylinder, }

C, Exhaust-pipe, leading to

D, Surface-condenser, composed of horizontal brass steam-tubes of about 2500 square feet surface in all.

The cold water circulating pump is concealed.

E, Air-pump, with piston-rod forming lower end of high-pressure steam piston-rod.

F, Bilge-pump, worked by arm projecting from air-pump piston-rod. (Feed-pump is similarly worked at other side of air-pump.)

G, Beam, or lever, of equal arms, to transmit motion and force from high-pressure piston-rod to low-pressure piston-rod; the latter rod being directly connected with the crank. (The lever and connecting-rod of the after engine are seen beyond.)

H, Spur-wheel gear, for driving the eccentrics.

K, Low-pressure slide-valve-rod.

L, High-pressure slide-valve-rod.

M, Reversing-gear, and donkey-engine to work it.

N, Wheel-trough, for containing inside-gear toothed wheel on after end of engine-shaft. This wheel drives a pinion on the forward end of the propeller-shaft. Multiple of gearing, $2\frac{1}{2}$.

Other arrangements of marine engines, illustrated by the Plates, are as follows:—

Direct-acting Inclined Paddle-wheel Engines (by Messrs. R. Napier & Sons, in the *Queen of the Orwell*), Plates $\frac{1}{1}$, $\frac{2}{2}$.

Oscillating Paddle-wheel Engines (by Messrs. John Penn & Son, in H.M.S. *Victoria and Albert*), Plate $\frac{3}{3}$.

Oscillating Paddle-wheel Engines (by Messrs. J. & G. Thomson, in the *Iona*), Plate $\frac{4}{4}$. The scale of Plate $\frac{4}{4}$ is four feet to an inch.

Oscillating Paddle-wheel Engines (by Messrs. Jones, Quiggin, & Co., in the *Hope*), Plates $\frac{5}{4}$, $\frac{6}{5}$.

Horizontal Double-trunk Screw-propeller Engines (by Messrs. John Penn & Son), Plate P, Figs. 1 and 2.

Horizontal Direct Screw-propeller Engines (by Messrs. Humphrys & Tennant), Plate P, Figs. 3 and 4.

47. *Nominal Horse-power* is a conventional mode of describing the *dimensions* of a steam-engine, for the convenience of makers and purchasers of engines, and bears no fixed relation to *indicated* or to *effective* horse-power.

The mode of computing nominal horse-power, established amongst *civil* manufacturers of steam-engines by the practice of Messrs. Boulton and Watt, is as follows:—

Assume the velocity of the piston to be 128 feet per minute \times cube root of length of stroke in feet;

Assume the mean effective pressure to be 7 lbs. on the square inch;

Then compute the horse-power from those assumed data, and the area of the piston; that is to say,

$$\begin{aligned} \text{Nominal H.-P.} &= 7 \times 128 \times \sqrt[3]{\text{stroke in feet}} \\ &\times \text{area of piston in square inches} \div 33,000 \\ &= \frac{\sqrt[3]{\text{stroke in feet}} \times \text{area piston in inches}}{47 \text{ nearly}} \\ &= \frac{\sqrt[3]{\text{stroke in feet}} \times \text{diam.}^2 \text{ of piston in inches}}{60} \end{aligned}$$

The indicated power of different engines usually exceeds the nominal power as computed by the above rule in proportions ranging in ordinary cases from 2 to 4, and in extreme cases from $1\frac{1}{2}$ to 5.

In the rule established by the Admiralty for computing the nominal horse-power of screw-propeller engines, the *real velocity* of the piston is taken into account; but the *fictitious effective pressure* of 7 lbs. on the square inch is assumed; consequently, by the Admiralty rule,

$$\begin{aligned} \text{Nominal H.P.} &= \text{velocity of piston in feet per minute} \\ &\times \text{area of piston in inches} \times 7 \div 33,000 \\ &= \frac{\text{velocity in feet per min.} \times \text{diam.}^2 \text{ in inches}}{6000} \end{aligned}$$

The indicated power of screw-propeller engines ranges from *once to three times*, and sometimes reaches *six times*, but is on

an average about *twice* the nominal power as computed by the Admiralty rule.

The Admiralty rule for the nominal horse-power of paddle-wheel engines is the same with the civil rule.

The preceding rules for computing the nominal power of engines are applicable to low-pressure engines alone. For high-pressure engines there is a customary rule proposed by Mr. Bourne, which consists in assuming the effective pressure to be 21 lbs. per square inch, the other data being the same as in the rule for low-pressure engines.

48. *Cylinders* are made of the toughest cast-iron that can be got. The thickness required for the sake of mere tenacity, to resist the internal pressure, might be calculated from the principles of the Third Division, Article 32, page 129, allowing *six* as a factor of safety; but in order that the cylinder may have that stiffness which is necessary to enable it to preserve its figure with great accuracy, it must be made many times thicker than is necessary for mere strength. The actual factors of safety of the cylinders of steam-engines, as they occur in practice, range from 30 to 40.

The bottom of a cylinder is sometimes cast in one piece with it, sometimes bolted on. The cylinder cover is bolted on. Care must be taken that the bolts have sufficient strength to withstand the pressure. The bottoms and covers of large cylinders are often made of the form of a segment of a sphere of large radius—in which case the two sides of the piston are made of the same figure, in order that space may not be lost in clearance. (See Plate $\frac{M}{7}$.)

The effect of the *jacket* has been fully considered in the preceding Section. The jacket ought to envelop not merely the body of the cylinder, but at least one end also, and, if possible, both ends, as in Plate $\frac{M}{7}$. Whether the cylinder is jacketed or not, it should always be clothed with an outer casing of felt and wood, if economy of heat is desired.

In *Double-cylinder Engines* it is favourable to the efficiency of the steam that the large and small cylinders, as in Plate $\frac{M}{7}$, should lie side by side in close contact; their pistons moving opposite ways at the same time, and either connected by means of a beam (as in the same Plate), or by driving a pair of opposite cranks; the last arrangement being the most favourable to the diminution of friction.

Treble-cylinder Engines differ from double-cylinder engines merely in having a pair of large cylinders for the continuation of the expansion, one at each side of the small cylinder, instead of one large cylinder. In the best form of treble-cylinder engine the piston of the central small cylinder drives one crank, and those of the two lateral large cylinders drive a pair of cranks pointing the opposite way to the middle crank. If half the work is performed in the middle cylinder, and the other half divided equally between the lateral cylinders, there is an exact balance of pressures on the shaft, and the friction of its bearings is simply that due to the weight resting on them.*

An *Oscillating Cylinder* is mounted on *gudgeons* or *trunnions*, generally near the middle of its length, on which it is capable of swaying to and fro through a small arc, so as to enable the piston-rod to follow the movements of the crank, to which

it is directly attached without the intervention of a connecting-rod. This construction is very advantageous in point of economy of space and weight. (See Plate $\frac{M}{7}$.)

The trunnions are hollow, and are connected by steam-tight joints, one with a steam-pipe leading from the boiler—the other with an exhaust-pipe leading to the condenser. The valve chest oscillates with the cylinder. Various arrangements are used to adapt the valve gearing to the oscillating motion of the cylinder and valve chest; one of the simplest being to communicate motion from the eccentric to a sliding rod on which is a cross-head of the form of an arc of a circle described about the axis of the trunnions when the valve is in its middle position, and having in it a slot of the same figure; in that slot is a slider attached to the end of a lever arm projecting from a rocking shaft carried by the cylinder; another arm projecting from that shaft moves the slide-valve rod.

In some American steamers, the place of an ordinary cylinder is supplied by a *sector* of a cylinder, in which a rectangular piston oscillates to and fro like a door on its hinge. In the position of the hinge is a rocking shaft, to which the piston is fixed; and by means of an arm projecting from one of the outer ends of that shaft and a connecting-rod, motion is communicated to the crank.

In *Rotatory Engines* the place of an ordinary cylinder is supplied by a vessel of the shape of a cylinder, a zone of a sphere, or some other solid of revolution, traversed along the direction of its axis by a shaft, which carries one or more revolving pistons.

In the *Disc Engine* the cylinder, or vessel which acts as a cylinder, is bounded laterally by a spherical zone, and endwise by a pair of cones, whose summits coincide with the centre of the sphere. The piston is a flat circular disc, fitting the interior of the spherical zone round its edge. There is a fixed partition in the cylinder, shaped like a sector of a circle; a radial slit in the disc fits this partition. The disc is fixed to a ball, being the joint on which it turns; and from that ball projects a rod, perpendicular to the plane of the disc. This rod acts in a manner as a crank pin; for its end fits into a round hole at the end of the crank, which is carried by the shaft, whose axis coincides in direction with the common axis of the spherical zone and of the two cones. By the disc and partition the cylinder is divided at each instant into four spaces, two of which are enlarging while the other two are contracting, as the crank revolves; the steam is admitted into the two former spaces, and discharged from the two latter spaces, by ports near the partition, and can be cut off if required by an expansion valve; thus the disc is made to take a sort of motion of *precession*, and the crank shaft is driven round. The angle between the shaft and crank pin is one-half of the angle between the sides of the two cones.

Rotatory engines and disc engines are used chiefly in small screw-steamers.

Amongst the parts and fittings of cylinders are, the *escape-valves*, one at each end, being valves opening outwards, and held shut by steel springs, for discharging water which may condense in the cylinder, or come in the liquid state from the boiler; the *grease-cock*, for lubricating the piston; *stuffing-boxes*, one for each piston-rod; the *ports*, or openings for the admission and exhaust of steam, with the *nozzles* or passages leading

* H.M.S. *Constance* has a pair of treble-cylinder engines of this class, by Messrs. Randolph, Elder, & Co.

to them. These last, as well as the steam-valves and valve-gear, will be considered further in a later Article.

49. Ordinary *pistons* with the common *metallic packing* are illustrated in Plate $\frac{N}{1}$. There are usually two, and sometimes three, rings of packing, each consisting of arcs of metal, built together so as to break joint, and pressed outwards against the interior of the cylinder by means of springs. Simpler arrangements are often used, such as one in which there is only a single packing ring divided at one point, and pressing against the sides of the cylinder by its own elasticity, which, as the ring is originally made of a radius a little larger than that of the cylinder, causes it to tend to expand. The gap at the point of division is sometimes filled by a tongue-piece mortised into the ends of the ring; sometimes by a small wedge-formed block, pressed outwards by a spring behind it.

Hemp is sometimes used as an elastic material behind metallic packing, to keep it pressed against the cylinder.

Metallic rings, or pieces of sheet brass, packed behind with hemp, are used also for the packing of stuffing-boxes.

50. *Piston Rods and Trunks*.—In ordinary engines each piston has but one rod, fitted at one end into a conical socket in the centre of the piston, and fixed by means of a gib and cotter, or a screw and nut. The piston-rod passes through a stuffing-box in the centre of the cylinder cover. (See Plates $\frac{N}{1}$, $\frac{M}{1}$.)

In some marine engines, two piston-rods, and in some four, are attached to one piston, and traverse a corresponding number of stuffing-boxes in the cylinder cover. These arrangements form part of peculiar systems of mechanism for connecting the piston with the crank. (See Plate $\frac{L}{1}$.)

A *trunk* is a tubular piston-rod, used to enable the connecting-rod to be jointed directly to the piston, or to a very short inner piston-rod, so as to save room in marine engines. The width of the trunk must be sufficient to give room for the lateral motion of the connecting-rod. A *double trunk* goes completely through the cylinder, which has a stuffing-box at each end. (See Plate P, Figs. 1 and 2.)

As to the exact determination of the *strength of piston-rods*, see the Third Division, Article 38, pages 131, 132. The piston-rod is to be regarded as a pillar fixed at one end and jointed at the other. In computing the stress on a piston-rod, the *greatest* effective pressure of the steam must be taken into account. The usual *factor of safety* is about 6 or 7; but in some cases it is as low as 5, and in others as high as 10. In engines of ordinary proportions, it is sufficient to multiply the greatest effective pressure of the steam in lbs. on the square inch by the area of the piston, and divide by 2500, to find the sectional area of the piston-rod.

51. The *Common Condenser* is a cast-iron vessel of any convenient shape, and strong enough to bear the atmospheric pressure from without, in which the waste steam from the cylinder is condensed by a shower of cold water from the *injection rose*. (See Plates $\frac{N}{1}$, $\frac{N}{2}$.)

The *capacity* of the condenser ranges from $\frac{1}{4}$ to $\frac{1}{2}$ of that of the cylinder.

The area of the *injection valve*, or *sluice*, by which the condensation water is introduced into the condenser from the sea, may be fixed by the following—

RULE.—Find the gross volume of condensation water required per minute by the rules of Article 45 of this Division; divide

that volume by 1620 feet; the quotient will be the required area.

In marine engines there is sometimes an injection valve leading from the ship's bilge into the condenser, which is opened only when the leakage of water into the ship threatens to become too great for the ordinary bilge pumps. On such occasions, the ordinary injection valve is closed.

The *Air Pump*, by which the water, air, and uncondensed steam are drawn from the condenser, and delivered into the *hot well*, or feed-water tank, when single-acting, is usually in engines with common condensers of a capacity from *one-fifth* to *one-sixth* of that of the cylinder; when the air pump is double-acting, it may of course be made one-half smaller. The valves through which it draws the water, steam, and air from the condenser, are called *foot valves*; those through which it discharges those fluids into the *hot well*, *delivery valves*. A single-acting air pump has *bucket valves* opening upwards in its piston. (See Plate $\frac{N}{1}$.) Flap valves, and other clacks of various forms, are used as air-pump valves. For an illustration of the circular India rubber flap valves, generally employed in engines of frequent stroke, see Plate $\frac{M}{1}$. The ratio of the area of the valve passages to that of the air pump piston ranges in different engines from $\frac{1}{2}$ to equality, being made greater as the speed of that piston is greater, so that the velocity of the fluids pumped may not in any case exceed about 10 or 12 feet per second.

The surplus water from the hot well, over and above that which is drawn away by the *feed pumps*, is discharged by marine engines into the sea.

The capacity of the *feed pumps* is regulated by the gross volume of feed-water required per revolution, as calculated by the rules of Article 45.

Sometimes the feed pumps are driven by a *donkey-engine*; and sometimes the boiler is fed by an *injector*, which will be again mentioned in Chapter III.

The condenser is provided with *blow-through valves*, communicating with the cylinder, usually shut, but capable of being occasionally opened, and with a *snifting valve* opening outwards to the atmosphere; through these valves steam can be blown to expel air from the cylinder and condenser before the engine is set to work.

A *surface condenser* consists generally of a great number of brass tubes, about $\frac{1}{2}$ inch in diameter, united at their ends by means of a pair of flat steam-tight vessels, or of two sets of radiating tubes, or in some other convenient manner. This set of tubes is inclosed in a casing, through which a sufficient quantity of cold water is driven. The steam being led by the exhaust pipe to one end of the set of tubes is condensed as it passes through them, and arrives in the state of liquid water at the other end of the apparatus, whence it is pumped away by the air pump. (See Plates $\frac{N}{1}$, $\frac{L}{1}$.) The capacity of the air pump of a surface-condenser, when single-acting, is about *one-eighth* of that of the cylinder.

The condensation-water is drawn from the sea, and driven through the condenser-casing by a pump called the *circulating pump*, or *cold water pump*, whose capacity depends on the gross volume of condensation-water required per revolution.

An ordinary value of the area of *tube-surface* in a surface condenser is about 10 square feet per nominal horse-power;

corresponding to from $2\frac{1}{2}$ to 5 square feet per indicated horsepower.

In a marine engine with a surface condenser, the loss of water by leakage, by blowing off at the safety valve, &c., is supplied by means of water distilled by suitable apparatus.

Amongst the ordinary fittings of every condenser is a *vacuum gauge*, for showing how much the pressure in the condenser is below that of the atmosphere.

Besides the pumps already mentioned, steam-vessels are provided with *bilge pumps* for discharging water from the hold.

52. *Steam Passages*.—The principle which ought to regulate the size of the steam-pipe, and of all nozzles, ports, and other passages by which the steam is *admitted* to the cylinder, has already been stated in Article 42 of this Division, viz., that the velocity of the steam should not be greater than 100 feet per second, supposing its density to be the same in the steam-pipe and in the cylinder during the admission.

To permit the ready escape of the steam during the back stroke, the exhaust-pipe should be about double the area of the steam-pipe.

The current of steam flowing along the steam-pipe can be diminished at will by the *throttle-valve*. When the throttle-valve is controlled by a governor, it is usually a disc-and-pivot valve, because that sort of valve is easily moved, the pressure of the steam on it being balanced.

A throttle-valve to be controlled by hand may be a disc-and-pivot valve, or an ordinary slide-valve moved by a screw, or a rotating slide-valve, or a conical valve moved by a screw; but the disc-and-pivot form is the most common.

The *stop-valve* is usually a double-beat valve moved by a screw, and is for completely closing the steam-pipe when required.

53. *Slide-valves*, on account of the simplicity of their action and the smoothness of their motion, are almost universally employed in Europe for the distribution of the steam in marine engines.

The *seat* of a steam-engine slide-valve consists usually of a very accurate plane surface, in which are oblong openings or *ports*. These are at least two in number; one communicating with each end of the cylinder.

The *long slide-valve* might also be called a sort of hollow or tubular *piston-valve*; for the back of the valve, which is semi-cylindrical, is made to move steam-tight at its top and bottom in the semi-cylindrical valve-chest, by means of two half-rings of metallic packing. The steam is admitted through the steam-pipe and throttle-valve to the middle part of the valve-chest. The two ends of the valve-chest communicate with the condenser, the lower end directly, and the upper end through the interior of the tubular part of the valve.

In many examples of long slide-valves, the two ends of the valve-chest have separate communications with the condenser. Such is the arrangement shown in Plates $\frac{M}{1}$, $\frac{N}{2}$.

The seat of the *short slide-valve*, besides the two steam-ports of the cylinder, has a third port, called the *exhaust-port*, between those two, which is the passage for the escape of the exhaust steam.

The short slide-valve is pressed against its seat, and the joint between it and its seat kept steam-tight, by the excess of the pressure of the steam in the valve-chest behind the valve, which comes from the boiler, above the pressure of the steam in the

interior of the valve, which communicates with the condenser or with the atmosphere, as the case may be.

In large engines, the amount of that difference of pressure, over the whole area of the face of the valve, would be unnecessarily great, causing too much work to be lost in overcoming friction. To diminish its amount is the object of the contrivance called the *equilibrium slide-valve*, in which the interior of the back of the valve-chest is a true plane, parallel to that of the valve-seat; and the back of the valve is provided with a flat brass packing-ring, which is pressed against the back of the valve-chest by springs. The amount of the pressure of the valve against its seat due to the pressure of the steam from behind, is the product of the intensity of that pressure into the excess of the area of the face of the valve above the area of the packing-ring at its back, and may be reduced to any required amount, how small soever, by making that ring large enough.

In a *double-cylinder engine* there are usually two slide-valves: one for the admission of steam into the high-pressure cylinder; the other for regulating the transfer of steam from the high-pressure cylinder to the low-pressure cylinder, and the exhaustion of the steam from the low-pressure cylinder to the condenser. (See Plate $\frac{M}{1}$.)

The "gridiron" form is sometimes given to the slide-valve in large engines: that is to say, each end of the cylinder has two or three parallel ports; and the valve is so formed as to connect the ports belonging to one end of the cylinder at the same time, with the valve-chest and the exhaust-port alternately.

54. *Eccentric*.—To produce the proper distribution of the steam by a slide-valve, the valve has a reciprocating motion of such a nature as to bring it to the ends of its stroke, being its greatest distances from its middle position, at periods intermediate between those at which the piston reaches the ends of its stroke. The *eccentric*, which is used to give that motion, is a circular disc carried by the shaft, with whose axis the centre of the disc does not coincide. It is equivalent to a crank whose length is equal to the *eccentric radius*; that is, the distance of the centre of the disc from the axis of the shaft; and being encircled with a hoop at one end of the *eccentric-rod*, it gives to that rod a reciprocating motion whose length of stroke, or *travel*, is the double of the eccentric radius. The eccentric-rod is either directly jointed to the slide-valve-rod, or connected with it by any convenient combination of levers and linkwork; and the joint in many cases consists of a notch, called the *gab* of the eccentric-rod, which takes hold of a pin.

By means of a suitable handle, the gab and pin can be disengaged and re-engaged, so as to throw the valve-motion "*out of gearing*" and "*into gearing*," and thus make the slide-valve stop and resume its motion when required.

In many engines a different contrivance is used, called the "*link-motion*," to be again referred to.

In many sets of marine engines, the eccentrics, instead of being on the engine-shaft, are on a smaller shaft driven by toothed gearing. Such is the case in the engines shown in Plates $\frac{1}{1}$, $\frac{M}{1}$.

To *reverse* the direction of rotation of the shaft of a steam-engine, the piston must be made to come to rest and then to move the reverse way, before completing a stroke, and the eccentric must assume that position relatively to the crank which is proper for working the slide-valve when the rotation

of the shaft is reversed. That position (or the position of *backward gear*) is somewhat less than half a circumference from the position of *forward gear*, measured round the shaft in the *direction of forward rotation*. To bring the eccentric, therefore, into backward gear, it is sufficient to cause it first to stand still while the shaft nearly finishes the first half-turn backwards, and then to accompany the shaft in its rotation.

In many marine engines, those objects are effected by having the eccentric *loose* on the shaft, and so counterpoised that its centre of gravity shall be in the axis of the shaft, but prevented from turning completely round by means of two shoulders, one of which holds it in the position of forward gear, and the other in that of backward gear; care being taken that the motion of the loose eccentric round the shaft shall be *forwards* to go from forward into backward gear, and *backwards* to go from backward into forward gear.

In one method of reversing an engine with a loose eccentric, the gab is disengaged from its pin, and the slide-valve shifted by hand if necessary. When the shaft has made part of a turn backwards the gab is re-engaged.

According to another mode of reversing by the loose eccentric, the eccentric, instead of standing still till the engine has turned back, is made, sometimes by a combination of wheel-work, and sometimes by the longitudinal motion of a shaft with a spiral rib or feather, to *overtake* or *outrun* the shaft while the engine is moving forward, until it reaches the position of reverse gearing; and the reversal of the motion of the engine follows. In this case the reversing gear is often driven by a donkey-engine, as in Plate $\frac{7}{2}$. The longitudinal motion of the eccentric-shaft is produced by a toothed wheel taking hold of circular projections which go round the shaft, and resemble in longitudinal section the teeth of a rack.

55. *Expansive Working by the Slide-valve.*—To understand how expansive working is produced by means of the slide-valve,



let the attention be fixed, in the first place, on one port only, represented in section by AB, Fig. 8. Let A be the *induction-side* of the port; that is to say, the side at which steam enters: and B the *eduction-side*; that is to say, the side at which steam escapes to the condenser, or to the air. Let CD be that part of the slide-valve which closes the port, AB; C being the *induction-edge*, and D the *eduction-edge*, of that part of the valve. Let motion of the slide-valve in the direction CD be called *forward*, and in the direction DC, *backward*.

Let the slide, as shown in the figure, be at the middle of its *stroke*, *throw*, or *travel*. The motion given to it by the eccentric is a back-and-forward motion, carrying the valve alternately before, and behind its middle position, to a distance called the *half-travel* or *half-throw* of the slide (which distance is equal to the eccentric radius).

If the valve is just large enough to cover the port and no more, the port is open for the admission of steam during the whole of the half-revolution which carries the valve before its middle position, and open for the exhaustion of steam during the whole of the half-revolution which carries the valve behind its middle position; the greatest width of opening of the port is equal to the half-travel of the valve; and if the eccentric is so

set that the middle position of the valve occurs exactly when the motion of the piston is reversed, the engine works with *full steam*; that is, without expansion.

To make the valve cut off the steam before the end of the stroke, it must be wider than the port, as exemplified in the figure. The valve being in its middle position, AC is called the *lap* (or *cover*) on the *induction-side*; BD the *lap* (or *cover*) on the *eduction-side*. The greatest openings of the port now become—

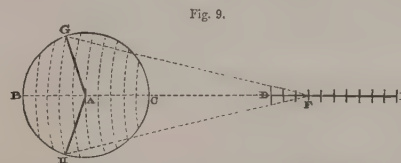
For admission, the half-travel less the lap on the induction-side;

For exhaustion, the half-travel less the lap on the eduction-side.

The eccentric must be set in *advance* of the position suitable for working with full steam, otherwise the admission of steam will not begin at the beginning of the stroke of the piston.

The following is the simplest system of rules for properly adjusting the lap of the slide-valve, and the advance of the eccentric:—

RULE I.—To find the positions of the crank corresponding to given points in the stroke of the piston—In a direct-acting engine, let A represent the axis of the shaft, B G C H the



circle described by the centre of the crank-pin. Draw the straight line, BACDE, in which lay off BD = CE = the length of the connecting-rod. Then DE will represent the whole length of stroke of the piston, and D and E will be the positions of the piston respectively corresponding to the "dead points," B and C, of the crank-pin. Let F represent any intermediate position of the piston; about F, with a radius equal to the length of the connecting-rod, describe a circular arc cutting the circle, BC, in G and H; these points will be the positions of the crank-pin corresponding to the position, F, of the piston; and the radii, AG and AH, will be the corresponding positions of the crank.

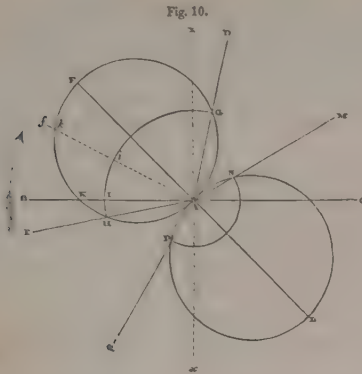
In the case of a *side-lever engine*, for the straight line there must be substituted a circular arc, to represent that described by the connecting-rod ends of the side-levers.

RULE II.—Given, the positions of the crank at the instants of admission and cut-off; to find the proper angular advance of the eccentric, and the proportion of the lap on the induction-side to the half-travel of the slide.*

In Fig. 10, let AB and AC be the positions of the crank at the beginning and end of the forward stroke; let the arrow show the direction of rotation; let Xx be perpendicular to BC; let AD be the position of the crank at the instant of cut-off, and AE its position at the instant of admission. (The admission is sometimes made to take place a little before the beginning of the stroke, in order to insure that there shall be a free entrance for the steam.) Draw AF, bisecting the angle EAD; AF will represent the position of the crank at the instant when the slide is at the *forward end* of its stroke; and FAX will be

* The method used in this and the following rules is that of Professor Dr. Zeuner, of the Swiss Federal Polytechnic School at Zürich, published in his treatise on Slide-valve Gearing, entitled "Die Schiebersteuerungen."

the *angular advance of the eccentric*; that is, the angle at which the eccentric radius should be set in advance of the position



which would bring the eccentric to the middle of its travel at the beginning of the stroke of the piston.

Lay off the distance AF to represent the half-travel; and on AF as a diameter describe the circle AHF FG , cutting AD in G , and AE in H ; then $\frac{AG}{AF} = \frac{AH}{AF}$ will be the *required ratio of lap at the induction-side to half-travel*; and $AG = AH$ will represent that lap, on the same scale on which AF represents the half-travel.

About A draw the circular arc, HG . Then if Af (cutting arc AHF in k , and arc HG in i) be the position of the crank at any instant, ik will represent the width of opening of the valve at that instant, on the same scale on which AF represents the half-travel.

On the same scale, IK represents the *width of opening of the valve at the beginning of the stroke*, sometimes called the "*lead of the slide*." Strictly speaking, this is the lead of the induction-edge of the slide only: the lead of the centre of the slide being AK ; that is, its distance from its middle position at the beginning of the forward stroke.

RULE III.—Given, the data and results of the preceding rule, and the position, AM , of the crank at the instant of release: to find the ratio of lap on the eduction-side to half-travel, and the position of the crank when cushioning begins—Produce FA to L , making $AL = AF$; on AL as a diameter draw a circle cutting AM in N : then $\frac{AN}{AL}$ will be the *required ratio of lap at eduction-side to half-travel*.

About A draw the circular arc NP , cutting the circle, AL , again in P ; join AP : then AP will be the *required position of the crank at the instant when cushioning begins*.

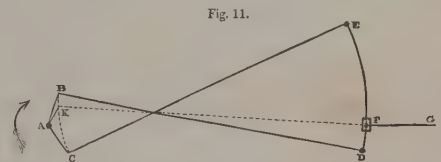
RULE IV.—Given, the data and results of Rule III., and the position, AQ , of the crank at the instant of cushioning: to find the ratio of lap at the eduction-side to half-travel, and the position of the crank at the instant of release—Produce FA as before; on $AL = FA$ as a diameter draw a circle cutting AQ in P : $\frac{AP}{AL}$ will be the *required ratio of lap at the eduction-side to half-travel*.

About A draw the circular arc PN , cutting the circle, AL , again in N ; join AN : AN will be the position of the crank at the instant of release.

RULE V.—Given, the angular advance of the eccentric, the half-travel of the slide, and the lap at both sides: to find the positions of the crank at the instants of admission, cut-off, release, and cushioning—Draw the straight lines, BAC and XAx , perpendicular to each other; and take B and C to represent the dead-points. Let the arrow denote the direction of rotation. Draw FAL , making the angle FAX = the angular advance of the eccentric; and make $AF = AL$ = half-travel. On AF and AL as diameters, draw circles. About A , with a radius equal to the lap at the induction-side, draw an arc cutting the circle on AF in H and G ; also, with a radius equal to the lap at the eduction-side, draw an arc cutting the circle on AL in N and P . Draw the straight lines, AHE , AGD , ANM , APQ . These will represent respectively the positions of the crank at the instants of *admission, cut-off, release, and cushioning*.

The preceding rules are to be applied *separately to the two ends of the cylinder*. The angular advance, and the half-travel, are of course the same for both ends; but the lap may be and often is different, whether at the induction-side or at the eduction-side; especially when the steam is expanded at different rates on the two sides of the piston, for the purpose explained in Article 45A of this Division.

56. The *Link-motion* is a contrivance for reversing the motion of the engine, and varying the rate of expansion at will, whose general principle is shown in Fig. 11, while examples of its application are given in Plate P. In Fig. 11, A represents the



axis of the engine-shaft, and the arrow, the direction of forward rotation. For each engine a pair of eccentrics are used; AB being the eccentric radius of the *forward eccentric*, and AC that of the *backward eccentric*. BD and CE are the two eccentric rods, connected with the two ends of the *link*, DE , which has a radius of curvature equal, or nearly equal, to the length of the rods. The link sometimes consists, as in Plate P, Fig. 1, of a pair of curved bars with a space or *slot* between them for a sliding block, and sometimes, as in Plate P, Fig. 3, of a single curved bar with a block sliding upon it. The sliding block or *stud*, F , is jointed to the end of the slide-valve rod, FG ; and the link is hung so as to be capable of being shifted to the extent of its length, by means of gearing worked either by hand or by a donkey-engine.

When the stud, F , is at the end D of the link, the motion of the slide-valve is produced wholly by the forward eccentric, B , and the engine is said to be in "*full forward gear*," and is driven ahead at the greatest speed. When the stud, F , is at the end E of the link, the motion of the slide-valve is produced wholly by the backward eccentric, C , and the engine is said to be in "*full backward gear*," and is driven astern at the greatest speed.

In intermediate positions of the stud, such as that shown in the figure, the motion of the slide-valve is produced by the joint action of the forward and backward eccentrics, and may be determined as follows: with a radius bearing the same pro-

portion to the distance, BC, that the radius of curvature of the link bears to its length, DE, draw the arc, BC. If the eccentric-rods are so placed (as in the figure) that when the eccentrics are inclined towards the link, the rods are crossed, make the arc, BC, convex towards the axis, A. If, on the contrary, the eccentric-rods are so placed as not to be crossed when the eccentrics are inclined towards the link, make the arc, BC, concave towards A. In that arc take a point, K, dividing it in the same proportion in which the stud, F, divides the link, DE. Then the motion of the stud, F, will be very nearly the same as if it were directly connected by a rod, KF, with a crank, AK. Consequently, from the *half-travel*, AK, and the angular advance, of that supposed crank, the motions of the slide-valve and their effects may be deduced by Rule V. of the preceding Article.

The use of the link-motion in producing high rates of expansion in a single-cylinder engine is limited by the fact, that an early cut-off involves a great angular advance and short travel, and that these produce an early release and cushioning, which may interfere with efficient working. One way of avoiding this difficulty is to have *separate induction and eduction ports* in the cylinder, with separate slide-valves, pairs of eccentrics, and link-motions, but the links shifted by one handle; when the eccentrics of the eduction-slide may be made so as to have no more angular advance and lap than are consistent with good working. This arrangement has the additional advantage (first pointed out by Dr. Joule) of not exposing the induction-ports to be cooled by the exhaust steam.

Another method is to have *double slides*,^o one on the back of another, of which great varieties have been contrived. A third, and the most common way, is to have a separate expansion-valve.

57. *Separate Expansion-valve*.—One form of expansion-valve consists of a double beat valve, whose spindle is hung from one arm of a lever: another arm of the lever has a roller at its end, upon which a suitably shaped cam, turning with the engine shaft, acts so as to lift the valve twice in each revolution, hold it open during the proper period for the admission of the steam, and then let it close. A series of such cams, suited to produce different rates of expansion, are fixed side by side on the shaft, and the lever arm which carries the roller is so made that it can be shifted sideways, and brought into gearing with that cam which produces the proper rate of expansion. In some cases the rate of expansion is adjusted by a single cam, shifting endways along a screw-shaped part of the eccentric-shaft under the action of the governor.

A separate *expansion slide-valve* is sometimes used to cut off the steam at an early period of the stroke, worked either by cams like those already mentioned (see Plate $\frac{N}{1}$) or by an eccentric *without angular advance*; in which latter case this valve is at its middle position when the piston is at either end of its stroke. A longitudinal section of such a valve, worked by cams, is shown in Plate $\frac{N}{2}$, Fig. No. 4; and an inside view

of the seat in Plate $\frac{N}{1}$. There are two, three, or more equal and similar oblong ports in the plate which forms the valve-seat, and which is usually the back or the side of the valve-chest of the ordinary slide-valve. In the sliding plate which forms the valve, are a set of openings equal and similar to the ports in the valve-seat; whence the valve is called a "*gridiron valve*." When the piston is at either end of its stroke, the openings in the valve are exactly opposite to the ports in the seat, which are then "*full open*." So soon as the valve has moved in either direction to a distance equal to the width of one of its openings, the ports are all closed, and the steam cut off.

This valve is suited for cutting off the steam at an early period of the stroke only. The point of cut-off being given, the following is the process for finding the requisite *proportion of breadth of openings to half-travel*. (See Fig. 12.)

Let AB represent the position of the crank at one end of the stroke, or dead-point, and AD = AB, its position at the instant of cut-off; from D let fall DE perpendicular to AB; then

$$\frac{\text{Breadth of openings}}{\text{Half-travel of valve}} = \frac{DE}{AB}$$

The rate of expansion may be varied by varying the travel of the expansion slide-valve; and this may be effected either by driving the valve by means of a link-motion, or by making the eccentric drive a slotted curved lever, having a sliding stud in it connected by a link with the expansion-valve rod, and capable of being slid to different distances from the fulcrum of the lever.

58. A *Governor* is not always a part of a marine engine; but in sea-going vessels it is materially conducive both to economy and to safety.

A governor for a marine engine ought to have all its moveable parts exactly balanced, so that gravity may have no effect upon their movements. It consists in general of the following essential parts:—

I. A *spindle* or axis, driven by the engine.

II. Two or more equal weights, symmetrically arranged, carried round with the rotation of the axis by means of jointed levers or rods, so connected together that although the weights are free to move from or towards the axis, their common centre of gravity always remains at a fixed point in the axis.

III. A spring or springs, pulling the weights towards the axis, and balancing their centrifugal force, when they revolve at the proper speed.

Rules for the centrifugal force of a single weight have been given in the First Division, Article 80, page 33. In the present case there is usually a more or less complex combination of weights; and the levers, rods, or other moveable parts, must be taken into account as well as the weights. Hence the following—

RULE.—To calculate the tension of the spring or springs corresponding to a given speed of rotation of the governor: Take the centrifugal force of each moveable part of the governor separately, by multiplying its weight by the perpendicular distance of its centre of gravity from the axis in *feet*, and by the square of the number of turns per minute, and dividing by 2935; then conceive the moveable parts of the governor to

Fig. 12.

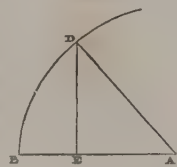
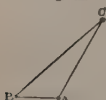


Fig. 1A.



* It is unnecessary here to give the theory of double slides in detail; but it may be explained that it depends mainly on the following principle:—In Fig. 11A, let A be the axis of the shaft, and AB and AC the eccentric-radii of two eccentrics which drive two slides, the first of which slides on a fixed seat and the second on the back of the first. Then the motion of those slides *relatively to each other* is the same as if the first were fixed and the second driven by an eccentric of a radius equal and parallel to BC.

undergo a small alteration of position; multiply the centrifugal force of each part by the ratio which the change in its distance from the axis bears to the corresponding change in the length of the springs; then add together the products; the sum will be the tension required.

IV. The moveable parts of the governor are to be so connected either with the throttle-valve, or (what is better) with the link-motion, or other gearing that regulates the expansion-valve, that any excess of speed shall diminish, and any deficiency of speed increase, the supply of steam.

59. *Guides for the Piston-rod* are very accurately straight surfaces, plane or cylindrical, but best plane, on which a block fixed to the head of the piston-rod slides, and which resist the tendency of the side-rods, or of the connecting-rod, when in an oblique position, to make the motion of the piston-rod deviate from a straight line.

Parallel Motions are jointed combinations of levers and link-work, designed to guide the motion of the piston-rod either exactly or approximately in a straight line, in order to avoid the friction which attends the use of straight guides. The levers and links of which a parallel motion consists are usually in pairs of equal and similar pieces, one at each side of the engine. The following are the rules for designing parallel motions which are of the most frequent use in marine engineering:—

RULE I.—Given (in Fig. 13) the line of motion, GD , of a piston-rod, the middle position of its head B , and the centre A of a lever which, in its middle position AD , is perpendicular to GD : to find the radius of the lever, so that the link connecting it with B shall deviate equally to the two sides of GD during the motion; also, the length of the link.

Make $DE = \frac{1}{2}$ stroke; join AE ; and perpendicular to it, draw EF cutting AD produced in F ; AF will be the required radius.

Join FB ; this will be the link.

RULE II.—Given, the data and results of Rule I.; also the point G where the middle position of a second lever connected with the same link cuts GD : to find the second lever, so that the two extreme positions of B shall lie in the same straight line, GBD , with the middle position.

Through G draw a straight line, $L GK$, perpendicular to GD ; produce FB till it cuts that line in L ; this point will be one end of the required second lever at mid-stroke, and FL will be the entire link. Then in DG lay off $DH = GB$; join AH , and produce it till it cuts $L GK$ in K ; this will be the centre for the second lever.

When the two extreme positions and the middle position of B lie in the straight line GD , the whole of its positions are near enough to that line for practical purposes.

RULE III.—Given (in Fig. 14) the middle positions AC and BE (parallel to each other) and the extreme positions AD , AD' , BF , BF' , of two levers centred at A and B respectively: required the radii of those levers, so that a link connecting them shall deviate in direction by equal angles alternately to the two sides of a perpendicular to AC and BE .

From B let fall BG perpendicular to AC (produced if necessary). Draw AH bisecting the angle CAD . Also, at the point G , lay off the angle $AGH = \frac{1}{2}EBF$; and let GH cut AH in H . Draw HK perpendicular to HA , and HL perpendicular to HG , cutting AG in K and L respectively. In BE lay off $BM = GL$. Join KM . Then will AK and BM be the required radii of the levers, and KM the link connecting them.

RULE IV.—In the link, KM , found by the preceding rule, to find the point whose middle and two extreme positions lie in one straight line; also, the length of stroke of that point.

Through H draw PHN perpendicular to AK and BM . The required length of stroke is $4PH$. The required point is R , where PHN cuts KM ; but as the acuteness of the angle of intersection causes this method of finding R to be wanting in precision, it is better to proceed as follows; draw the straight line AB , cutting NP in Q ; then lay off $PR = NQ$: R will be the required point.

RULE V.—Given (in Fig. 15) the main centre A , the middle position of the main lever AF , the piston-rod head B , and its length of stroke; the radius, AF , of the lever, and the main link, FB , having been found by Rule I. Let the figure represent those parts at mid-stroke; and let it be required to construct a parallel motion consisting of a parallelogram, $CEDF$ (in which $CE = FD$ is called the *parallel bar*, and $DE = FC$ the *back link*) and a radius lever, or *bridle*, HE , jointed to the angle E of the parallelogram.

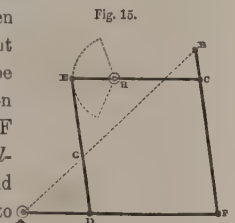
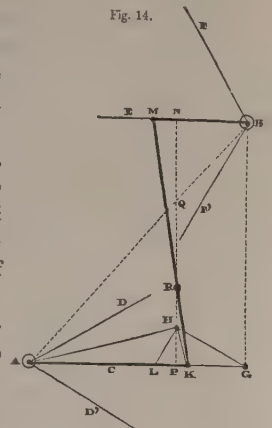
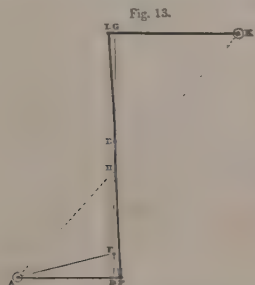
Draw the straight line AB , cutting the back link DE in G ; then by Rule II. find the lever HE , such that the middle and extreme positions of G shall lie in one straight line.

(The point G shows where a pump-rod may, if convenient, be jointed to the back link).

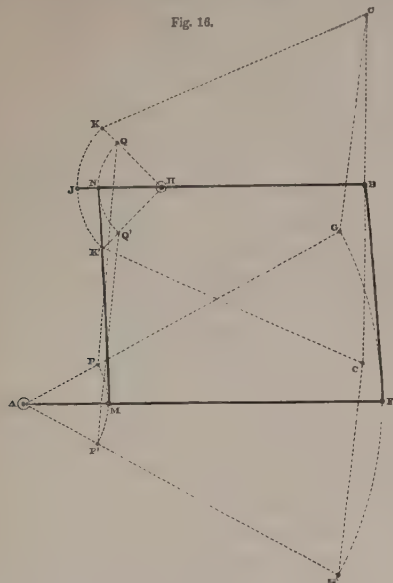
RULE VI.—Given (in Fig. 16), the main centre, A , and the main lever, AF , and main link, FB , found by Rule I. Let B be the middle position, and C and C' the two extreme positions, of the point where the parallel bar is jointed to the main link, whether at the piston-rod head or at some other point. Let AG and AG' be the two extreme positions of the main lever. Let H be a convenient point in the straight line, BH , parallel to FA , chosen as the centre for the radius lever; and let HJ be a convenient length chosen for the radius of that lever; J being the middle position of the point where it is jointed to the parallel bar.

Required, the points where the back link ought to be jointed to the main lever, AF , and the radius lever, HJ , respectively.

About H , with the radius, HJ , describe a circular arc; then take the length, BJ , of the parallel bar in the compasses, and lay off the same length from C and C' respectively, to K and K' in that arc: CK and $C'K'$ will be the extreme positions



of the parallel bar. Join HK, HK': these will be the extreme positions of the radius lever. Then, by Rule III., find the ends,



M and N, of the back link, MN, which is to connect the radius lever with the main lever.

The two extreme positions of the back link are marked PQ, P'Q'.

If it be desired to joint a pump-rod to the back link, the proper point for that purpose may be found by Rule IV.

The kind of parallel motion referred to in Rule VI. is exemplified in Plate $\frac{N}{1}$.

60. *Balancing of Engines.*—All the moving parts of an engine ought to be, as far as practicable, balanced; and the balance should be not merely a *standing balance*, or balance of weights, but also a *running balance*, or balance of reactions when the engine is in motion.

To insure a standing balance, it is sufficient that the resultant of the weight of each moving piece that turns upon an axis, together with any other weights with which that piece is loaded, should always pass through that axis. In the case of a shaft, this is effected by the aid of counterpoises, rigidly connected with the shaft in such positions as to balance the cranks and eccentrics; and in adjusting such counterpoises, half the weight of each connecting-rod may be regarded as a load applied to the crank-pin. In a side-lever engine, the weight of the side-levers, together with that of the pistons, rods, cross-heads, &c., with which they are loaded, and that of the cross-tail and half the connecting-rod, should be balanced upon the main centre.

In single-cylindrical vertical, or nearly vertical, direct-acting engines, it is considered better to balance the weight of the piston, piston-rod, and half the connecting-rod, by the pressure of steam (as explained in Article 45A of this Division), than by means of counterpoises; because counterpoises would derange the running balance.

All parts of valve-gearing require to be accurately counterpoised.

The reactions of the parts of an engine when in motion are

usually of two kinds: reactions of revolving parts, or *centrifugal forces*, explained in the First Division, Article 81; and reactions of reciprocating or oscillating parts, explained in the First Division, Article 84.

To insure a running balance of centrifugal force about a given axis, it is necessary, in the first place, that there should be a standing balance of the masses that turn about the axis; and, moreover, that there should be a balance of "centrifugal couples." Suppose a weight to be counterpoised by means of another weight, revolving in the same plane; then the centrifugal forces are not only equal and opposite, but directly opposed; and they balance each other completely. But if the weights, though still statically balanced upon the axis, revolve in different planes, then their centrifugal forces, though equal and opposite, are no longer directly opposed; and they constitute a couple, whose arm or lever is the distance between the planes of revolution, and which tends to displace the axis into a new direction. All counterpoises upon shafts, therefore, should be so placed as either to give rise to no centrifugal couples (which is the case when each counterpoise revolves in the same plane with the mass that it balances); or if that be impracticable, the counterpoises should be so arranged that the centrifugal couples may balance each other.

The reactions of reciprocating parts in a side-lever engine, or in a treble-cylindrical direct-acting engine, are balanced by the same arrangement which produces a balance of weights. But in single-cylindrical direct-acting engines, the reactions of the piston, piston-rod, and part of the connecting-rod cannot be exactly balanced without introducing unbalanced centrifugal forces; that is to say, if the mass of those reciprocating parts be conceived to be collected at the crank-pin, and a pair of counterpoises be fixed at the opposite side of the shaft sufficient to balance the crank loaded with that supposed additional weight, the centrifugal force of those counterpoises balances the reactions accurately when it acts in the direction of the stroke; but when that centrifugal force acts transversely to the stroke, it is itself unbalanced.

In screw-propeller engines, unbalanced horizontal reactions, producing thwartship vibrations, are much more prejudicial than unbalanced vertical reactions. Hence in such engines with vertical, or nearly vertical cylinders, the reactions of the reciprocating parts should be left unbalanced; but in those with horizontal, or nearly horizontal cylinders, those reactions should be balanced, as already described, by means of counterpoises, although the weights and vertical reactions of those counterpoises are thereby left unbalanced.*

61. *Bearing Surfaces.*—As to the bearings of paddle and propeller shafts, see Article 18 of this Division. The following remarks apply to the other bearings of the engine. The bearing surfaces of the moving parts of machinery should not be loaded beyond a certain intensity of pressure, lest the pressure force out the unguent, and make the surfaces grind each other and become overheated. That limiting pressure becomes the less, the greater the speed at which the surfaces slide over each other. The fulfilment of this condition requires that every such bearing surface should have at least a certain area, depending on the amount of the load and the speed of the sliding motion;

* This arrangement is due to Mr. John Bourne.

and it is always better to run the risk of making the bearing surface too large, than too small.

In the case of a journal, it is difficult to define exactly the extent of bearing surface; because a well-made bush for a journal never grips the journal tightly, but fits it easily; so that the actual bearing surface is a strip of the length of the journal, but of a breadth much less than the diameter of the journal. In practical rules, however, relating to this subject, it is assumed that the actual bearing surface bears some constant, though unknown, proportion to the *product of the length and diameter of the journal*; which product is what is meant when the *area of a journal-bearing* is spoken of; and the *pressure per square inch* means the load divided by that area.

The rule given by Mr. Bourne for the limiting pressure per square inch on a bearing is as follows:—

RULE I.—Add 50 to the *velocity of rubbing in feet per minute*, and divide 70,000 by the sum; with this qualification, that how slow soever the motion, the pressure should in no case exceed 1200 lbs. on the square inch.

This rule agrees with successful practice when the velocity of rubbing is moderate—say, up to 60 feet per minute, or thereabouts; but at high speeds, such as from 150 to 300 feet per minute, it gives pressures exceeding the limits allowed in the best practice of locomotive engineers. The following rule agrees nearly with Mr. Bourne's at speeds not exceeding about 50 feet per minute, and at higher speeds gives gradually less limits of pressure, so as to agree with locomotive engineering practice at 300 feet per minute:—

RULE I.A.—Add 20 to the *velocity of rubbing in feet per minute*, and divide 44,800 by the sum: the quotient will be the *pressure in lbs. on the square inch*; with the qualification, as before, that the pressure is in no case to exceed 1200 lbs. on the square inch.

RULE II.—When the pressure corresponding to the velocity has been computed, divide the load by that pressure to find the *area of bearing* required. The diameter of a journal is fixed by conditions of strength; therefore, divide the area by the diameter, and the quotient will be the *length of bearing* required.

In the ordinary practice of marine engineers, it is common to make the length of each journal about once and a quarter of its diameter.

62. Strength of Mechanism and Framing.—The properties of the materials of which the mechanism and framing of engines are made, have been stated in the Fourth Division, Chapter I.; and the principles of the strength of materials, with rules for their application to pieces loaded in different ways, have been given in the Third Division, Chapter I.

In applying those principles to marine engines, care must be taken to consider all the variations which the forces acting amongst the pieces of the mechanism undergo, whether in magnitude or in direction, and to take into account that condition of those forces in which the stress produced by them is the most severe; and in particular, regard must be had to the fact, that the direction of the stress upon almost every reciprocating piece is periodically reversed; for example, the piston-rods, side-rods, connecting-rods, &c., are exposed to tension and thrust alternately; and the side-levers, cross-heads, cross-tails, &c., are bent alternately in opposite directions.

The framework by which a moving piece is held or supported, exerts upon that piece a force or forces sufficient to prevent it from being dislodged from its proper bearings, and must be made sufficiently strong to bear with safety all the forces exerted by other bodies upon the moving pieces which it carries.

For example, in a side-lever engine, the principal parts of the framework are, the sole or base, the plumber-blocks for alternately supporting and holding down the main centre of the lever, and the pillars and stays for alternately supporting and holding down the shaft. At one end of the base, the cylinder must be fixed down to it by bolts capable of safely resisting an upward pull equal to the greatest effort on the piston. At the other end, the bearings of the shaft must be held down with equal firmness, and also supported in a manner sufficient to bear at once the greatest downward pull of the connecting-rod and the weight of the shaft. The supports of the main centre must be strong enough to bear the forces acting upon it, which consist of the weight of the side-levers and their load acting downwards, and a force equal to *twice* the greatest effort on the piston, acting alternately downwards and upwards. The base itself must possess transverse strength sufficient to bear safely the tendency of the forces applied to its ends and middle to break it across, producing a *moment of flexure* (Division First, Article 45) at each instant, equal and opposite to that which acts on the beam.

In a direct-acting engine, the principal parts of the frame are the pillars or rods by which the cylinder and the shaft are kept in their proper relative positions, and which have to resist a pull and a thrust alternately.

In designing the framing of engines, it must further be borne in mind that there is always a moment tending to *overturn* the framing, equal and opposite to that exerted by the connecting-rods upon the cranks; and that moment has to be resisted by the *holding down bolts* which connect the framing with the keelsons on which it stands.

In fixing the dimensions of all *fastenings* in engines, regard should be had to the principles of the First Division, Article 60.

Rules have already been given for the diameters of *shafts* in Article 11 of this Division, and for the diameters of *piston-rods* in Article 50. When those diameters have been determined, the dimensions of various other parts may be deduced from them. For example, the strength of the connecting-rod, and the collective strength of the side-rods in lever-engines, ought to be equal to that of the piston-rod; taking into account the fact, that connecting-rods and side-rods are jointed at both ends, and piston-rods at one end only; and the strength of a crank against bending ought to be equal to that of the shaft to which it belongs against twisting.

The same *proportionate dimensions* are suitable for all *similar engines*, large and small, in which the greatest effective pressure of the steam is the same; and in similar and equal engines with different pressures of steam, the diameter of the piston-rod should vary as the square root of the greatest effective pressure, and that of the shaft as the cube root.

The following statement of the proportionate dimensions of some of the principal parts of an engine, is based partly on the rules deduced by Mr. Bourne from the practice of Messrs.

		Diameter of Shaft-journal multiplied by
Piston-rod :	elliptical part in the piston, length,.....	2.00
"	" " greater diameter,.....	1.40
"	" " lesser diameter,.....	1.15
"	elliptical part in the cross-head, greater diameter,....	0.95
"	" " lesser diameter,.....	0.90
"	depth of cutter through piston,.....	0.85
"	thickness of cutter through piston,	0.85
Connecting-rod :	diameter at ends,.....	1.00
"	" at middle,.....	1.20
"	breadth of butt,.....	1.56
"	thickness of butt,.....	1.25
"	thickness of strap at cutter,.....	0.43
"	" elsewhere,	0.32
"	distance of cutter from end of strap,... ..	0.48
"	depth of gibs and cutter,.....	1.10
"	thickness of gibs and cutter,.....	0.30
Crank-pin journal :	diameter,.....	1.40
"	length,.....	1.60
Crank :	thickness of small eye,.....	0.63
"	breadth of small eye,.....	1.87
"	web : thickness produced to centre of crank-pin,.....	1.10
"	" breadth produced to centre of crank-pin,.....	1.60
		Diameter of Shaft-journal multiplied by
Crank :	web : thickness produced to centre of shaft,.....	0.75
"	" " breadth produced to centre of shaft,.....	1.50
"	thickness of large eye,.....	0.45
"	breadth of large eye,.....	1.75

than the propeller-shaft, in the ratio expressed by the "multiple of gearing;" that is to say, its diameter must be greater in the ratio of the cube root of that multiple. Inside gearing is preferable to outside gearing, on account of the greater smoothness of the motion. For the same reason, the teeth are often stepped. To find the proper thickness for a tooth, divide the greatest twisting moment in foot-pounds on the propeller-shaft by the radius of the pitch-circle of the pinion in feet, and by 1500: the square root of the quotient is the required thickness in inches.

The object in employing a geared engine in any given vessel, notwithstanding its being larger, heavier, and more expensive than a direct-acting engine, is mainly to lessen the waste of steam through clearance. The dimensions of the steam-valves, ports, and passages depend on the quantity and pressure of the steam used, and must therefore be the same for an engine of a given power, whether the revolution is rapid and the cylinder small, or the revolution slow and the cylinder large. Hence the waste of steam of a given pressure in an engine of a given power, which arises from the filling of ports and passages, depends mainly on the frequency with which they are filled, and is greater in the engine with the rapid revolution and small cylinder. Such is the case especially in engines working at a high rate of expansion.

The question between saving of steam and saving of cost, weight, and bulk of engines, is one for the judgment of the engineer in each particular case. (See page 292.)

OF MARINE BOILERS AND FURNACES.

REFERENCE TO BOILER IN PLATE $\frac{N}{3}$.

The boiler represented in the Plate is one of those of the *Arabia*, made by Messrs. Robert Napier & Sons, and belongs to the class of ordinary multitubular marine boilers. Fig. 1 is a longitudinal section of a boiler, and of one-half of a chimney. Fig. 2 is a transverse section of one-half of a boiler, the other half being similar. The boilers of the *Persia* are of similar construction, but different in dimensions and number; their general arrangement is shown in Plates $\frac{A}{2}$ and $\frac{A}{3}$.

- | | | | |
|----------------|--------------------------------|----------------|---------------------------|
| A | Furnaces. | F | Water-space stays. |
| A ¹ | Fire-bars or grate. | G | Uptake. |
| A ² | Furnace-doors. | H | Steam-chest. |
| B | Ash-pits. | J | " " separate. |
| B ¹ | Ash-pit dampers. | J ¹ | Steam-chest door. |
| C | Fire-box or flame-chamber. | J ² | Steam-pipe. |
| D | Tubes. | K | Safety-valve-chest seat. |
| D ¹ | Tube-ferules. | L | Funnel-seat. |
| D ² | " plates. | M | Connection or stop valve. |
| D ³ | " plate stays. | N | Vacuum-valve. |
| D ⁴ | " doors. | O | Water-gauge and cocks. |
| E | Horizontal and vertical stays. | P | Brine-pipe rose. |

- | | |
|---|-------------------------------|
| P ¹ Brine-valve. | T Stoke-hole. |
| P ² Feed-valve. | U Firing or stoking platform. |
| P ³ Brine-valve and feed-valve gear. | V Main deck. |
| Q Feed cock and valve. | W Coal deck, of iron. |
| R Priming-grease-cock. | X Sides of ship. |
| S Blow-off pipe. | Y Keelsons. |

DIMENSIONS AND NUMBER OF BOILERS OF THE ARABIA AND PERSIA.		
BOILERS.	ARABIA.	PERSIA.
No. 1.		
" 2.		
" 3.		
" 4.		
" 5.		
" 6.		
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" 97.		
" 98.		
" 99.		
" 100.		

33 Kind of boilers, and number.....	Tubular, 4,	... Tubular, 8
34 Furnaces, total number.....	24	... 40
35 Length and breadth of fire-bar surface, 8 ft. 4 in. x 3 ft. 2½ in....		6 ft. 6 in. x 2 ft. 9 in.
36 Area of fire-bar or grate surface.....	642 sq. ft.	715 sq. ft.
37 Area of heating surface.....	16948	" 21900 "
38 Length of firing or stoking spaces....	12 ft. 0 in.	... 12 ft. 0 in.
39 Boilers, how fired.....	Athwartship.	... Athwartship.
40 Position of boilers relative to engines, 2 afore, 2 abaft....	2 afore, 2 abaft....	4 afore, 4 abaft.

REFERENCE TO BOILER IN PLATE $\frac{L}{T}$.

The boiler represented is multitubular, with a superheating apparatus, and made by Messrs. Maudslay, Son, & Field.

- | | |
|--------------------------------------|--------------------------------------|
| A, Boiler. | by tubes which traverse the |
| B, B, Steam-chests. | chimney. |
| C, Safety-valve chest. | F, Stop-valve for superheated steam. |
| D, Supply-valve of superheater. | G, Stop-valve for common steam. |
| E, E, Superheater, consisting of two | H, Steam-pipe, leading to engines. |
| chambers connected together | |

64. The determination of the *principal dimensions of the Furnaces and Boilers* for a steam-vessel depends mainly upon the following principles:—

I. The greatest intended available heat, or rate of expenditure of heat upon the steam, is to be computed in units of work, by dividing the greatest indicated power required in units of work per unit of time (say in foot-pounds per hour) by the probable efficiency of the engine; or otherwise, by multiplying the pressure equivalent to the rate of expenditure of heat by the total cylinder-capacity, and by twice the number of revolutions per hour. (See Article 41 of this Division.)

EXAMPLE.—*First Method.*

Probable indicated horse-power at full speed,.....	743
× foot-lbs. per hour in an indicated horse-power,.....	1,980,000
Indicated power in foot-lbs. per hour,.....	1,471,140,000
The above, divided by the probable efficiency of the steam, 0.12, gives for the available heat required, in foot-lbs. per hour,.....	12,259,500,000

Second Method.

Estimated pressure equivalent to rate of expenditure of heat on steam,.....	108½	{ lbs. on the sq. in.
× estimated total cylinder-capacity in prisms of one foot × 1 inch × 1 inch,.....	89033	
	4,228,575	product.
× twice the number of revolutions per hour,.....	2899.2	
Available heat required in foot-lbs. per hour,.....	12,259,484,650	

II. The *available heat of combustion of one pound of fuel* is to be estimated by multiplying the *total heat of combustion of one pound of fuel* by the *efficiency of the furnace*.

A. The *total heat of combustion* of one pound of good marine steam-coal may be estimated at from 9,000,000 to 10,000,000 foot-pounds; and that of the very best quality at 12,000,000.

Inferior qualities range down to about two-thirds of the above estimates.

B. The *efficiency of the furnace* may be roughly estimated as follows:—Divide the intended number of square feet of heating surface per pound of fuel per hour by the same number + 0.5: eleven-twelfths of the quotient will be the probable efficiency of the furnace, nearly. The following are examples:—

	Square feet heating surface per lb. fuel per hour.	Efficiency of Furnace.	Available heat per lb. coal, if total heat is 10,000,000 ft.-lbs.
Small value for marine boilers,.....	0.50	0.46	4,600,000 ft.-lbs.
	0.75	0.55	5,500,000 "
	1.00	0.61	6,100,000 "
Ordinary values in marine boilers, {	1.25	0.65	6,500,000 "
	1.50	0.69	6,900,000 "
	2.00	0.73	7,300,000 "
Water-tube and cellular boilers, ... {	3.00	0.79	7,900,000 "
	6.00	0.84	8,400,000 "

The most common values of the available heat of a pound of good steam-coal in marine boilers are from 5,000,000 to 6,000,000 foot-pounds.

These estimates allow nothing for waste of fuel through bad firing. This, according to the degree of unskilfulness, may amount to from 20 to 50 per cent. of the heat that would otherwise be available.

III. To find the probable greatest *rate of consumption of fuel*: divide the available heat per hour by the available heat of combustion of one pound of fuel.

EXAMPLE.—Available heat per hour, 12,260,000,000 ÷ available heat of combustion per lb. coal, say 5,500,000 = 2229 lbs., probable consumption of fuel per hour.

IV. To find the proper area of heating surface: multiply the rate of consumption of fuel in pounds per hour by the intended

area of heating surface to each pound of fuel per hour (that is, usually, from $\frac{3}{4}$ to $1\frac{1}{2}$ square foot).

V. To find the proper area of fire-grate: divide the rate of consumption of fuel in pounds per hour by the weight of fuel in pounds to be burnt per hour on each square foot of grate. This quantity ranges, in ordinary boilers, from 16 to 24 lbs.; and the latter limit may be considered a suitable rate for the most rapid combustion and highest speed, provided air is admitted above the fuel to burn its gaseous constituents.

On some grates the rate of combustion is as low as from 6 to 12 lbs. per square foot per hour; but these require very careful management to prevent waste of heat through the admission of an excessive quantity of air.

VI. The total *sectional area of flues* or of *tubes* is from *one-fifth* to *one-seventh* of that of the fire-grate; the area of the *chimney*, about *one-tenth* of that of the grate.

VII. The capacity of a marine boiler (exclusive of furnace-room) is equal to the heating-surface multiplied by about 1 foot for flue boilers, or 0.625 foot for multitubular boilers. To this has to be added the capacity of furnaces, flues, and tubes, all contained within the boiler, which may be estimated as nearly equal to the area of fire-grate multiplied by from 6 to 8 feet; and of the total capacity, about one-fifth is steam-space, and the rest partly water-space and partly furnace-space.

65. The *Principal Parts and Appendages of a Furnace* are,

I. The *furnace proper*, being the space where the solid constituents of the fuel, and the whole or part of its gaseous constituents, are burned. In almost all marine boilers, the furnace is wholly within the boiler, and surrounded by water-spaces.

II. The *grate*, being that part of the bottom of the furnace proper which is composed of alternate bars and spaces, to support the fuel and admit air.

III. The *hearth* is a floor of fire-brick, on which, instead of on a grate, the fuel is burned in some furnaces; and on such a hearth or floor liquid fuel, such as petroleum or coal-tar, may be burned, by supplying it in a thin stream.

IV. The *dead-plate*, or *dumb-plate*, being that part of the bottom of the furnace proper, close to the doors, which consists of an iron plate, without bars and spaces. Many furnaces are without a dead-plate.

V. The *mouth-piece*, being the passage through which fuel is introduced, and sometimes also air. The bottom of the mouth-piece is the dead-plate. In many furnaces there is a mere doorway, and no mouth-piece; and such is the case in the furnace shown in Plate $\frac{N}{3}$.

VI. The *fire-door*, which closes the mouth-piece or doorway, and which may or may not have openings and valves in it to admit air.

VII. The *furnace-front*, above and on either side of the fire-door; and, in marine boilers, usually containing a water-space.

VIII. The *ash-pit*, being the space below the grate into which the ashes fall, and through which, in most cases, a great part of the supply of air is admitted. In marine boilers it is usually surrounded by water-space, like the furnace.

IX. The *ash-pit door*, used in some furnaces to regulate the admission of air through the ash-pit.

X. The *bridge*, being a low vertical partition at one end of the furnace (usually the back) over which the flame passes on its way to the flues or chimney. This is what is meant when

"the bridge" is spoken of without qualification; but the word *bridge* is also applied to any partition having a passage for flame or hot gas above or below it. Bridges are often built of fire-brick; often also made of plate iron, and hollow, so as to contain water within, and form part of the water-space of the boiler—they are then called *water-bridges*. The top of a water-bridge ought to slope or curve upwards towards the ends, to admit of the rapid escape of the bubbles of steam which form on its internal surface. Sometimes a water-bridge projects downwards from a part of the boiler above the furnace, leaving a passage below for flame—it is then called a *hanging-bridge*. A water-bridge with passages for flame both above and below is called a *mid-feather*.

XI. The *flame chamber*, being the space in which the combustion of the inflammable gases is or ought to be completed.

XII. *Air-passages*, of various constructions and in various situations, and with or without valves, to admit air for the combustion of the fuel, whether forced in by atmospheric pressure or by a blowing machine. They are often in the furnace-door; and then they have a collective area of about $\frac{1}{36}$ of that of the fire-grate.

XIII. *Flues and Tubes*, being passages traversed by the hot gas on its way from the fire to the chimney. They may be either horizontal, vertical, or inclined. Tubes for marine boilers are from 2 to 4 inches diameter.

XIV. The *chimney*, at the foot of which is sometimes a chamber called the *smoke-box*, or *uptake*, in which the various flues terminate.

XV. *Blowing apparatus*, used in order to produce a draught, whether by forcing air into the furnace by means of a fan, or by driving the gases out of the chimney by means of a blast-pipe.

XVI. *Dampers*, being valves placed in the chimney, flues, tubes, or air-passages, to regulate the draught and rate of combustion.

No one furnace possesses *all* the parts and appendages above enumerated; for some of them are substitutes for others, and some are only employed in furnaces of particular kinds.

66. The *Principal Parts and Appendages of a Boiler are*,

I. The *shell*, or external boundary of the boiler, for which the usual material is iron, although sheet copper is sometimes employed. The figures usually employed for the shells of boilers are, the spherical, the cylindrical, and the plane, and combinations of those three figures. The shells of ordinary marine boilers are of irregular shapes, adapted to the space in the ship which they are to occupy, and approximating more or less to rectangular figures, rounded at the corners and arched at the top. In some peculiar boilers, the shell is a vertical cylinder, or a cluster of vertical tubes connected by means of horizontal tubes (as in Mr. Craddock's boiler); or a set of square tubes or cells (as in Mr. J. M. Rowan's boiler); or a single spiral tube (as in Mr. Perkins' boiler). Tubes which thus contain water internally are called *water-tubes*, to distinguish them from tubes for transmitting flame. They ought to be vertical, or nearly so, that the bubbles of steam may readily escape from them. In the "Dundonald" boiler,* the water-tubes, which furnish the greater part of the heating surface, stand within a flame-chamber,

which is itself contained within the outer shell of the boiler and surrounded by water.

II. The *steam-chest*, or *dome*, being a part of the shell which usually rises above the level of the rest of the boiler, so as to provide a space in which the steam, before being conducted to the engine, may deposit any particles of spray that it may have carried up from the water. It is usually cylindrical, with a hemispherical, segmental, or flat top; and its form is much varied. It is advantageous that the steam-chest should be traversed by a flue, or by the uptake, in order to dry or slightly superheat the steam.

III. A *tube-plate* is a plate perforated with holes, into which the ends of a set of tubes are fixed. Each set of tubes requires a pair of tube-plates, one for each end of the tubes.

IV. The *man-hole* is a circular or oval orifice in any convenient position, large enough to admit a man to the interior of the boiler to cleanse or repair it. The entrance to the man-hole usually consists of a short cylinder having a flange surrounding its upper end, to which the cover is bolted, when the cover opens outwards. The bolts must be capable of safely bearing the pressure of the steam against the cover. Sometimes the cover opens inwards, and then it is kept shut by the pressure of the steam; but to prevent its being dislodged from its seat, it is held by bolts and nuts to cross bars outside the man-hole. The cover should fit its seat very accurately.

V. *Mud-holes* are orifices at or near the lowest part of a boiler, which are opened occasionally for the discharge of sediment.

VI. The *feed-apparatus* is usually supplied by a pump or pumps worked by the engine when in motion; the surplus water which comes from the feed-pump being discharged through a valve loaded with a pressure greater than that in the boiler.

VII. The *blow-off apparatus* consists, in fresh-water boilers, simply of a large cock at the bottom of the boiler, which is opened occasionally to cleanse the boiler, by emptying it completely of sediment and muddy water. In boilers fed with salt water, a similar cock is opened at regular intervals to discharge brine, and so prevent salt from collecting in the boiler. Another blow-off cock is sometimes so placed as to discharge occasionally the *scum*, consisting of crystals of salt, which collects on the surface of the water: this is called the *surface blow*.

As a substitute for the common blow-off apparatus, Messrs. Maudslay introduced *brine-pumps*, which draw off a fixed quantity of brine from the bottom of the boiler at each stroke of the engine. (See Article 45.)

The hot brine is, or ought to be, passed through a set of tubes, surrounded by a casing, through which the feed-water passes on its way to the boiler; the currents of the brine and of the feed-water flowing in opposite directions. By means of this apparatus, called the *refrigerator*, the greater part of the heat which would otherwise be wasted with the brine is saved by being transferred to the feed-water.

VIII. The *sediment collector*, used in some marine boilers, is a funnel shaped like an inverted cone, and placed within the boiler so that its mouth is somewhat above the water level. It communicates with the rest of the boiler through triangular slits near its upper edge. In the boiler generally, there is a continual boiling up of steam, which keeps crystals of salt and other solid particles for a time near the surface of the water. Within the cone there is comparatively still water, so that the

* The Dundonald boiler, modified by Mr. J. R. Napier, has given very economical results in the S. S. *Lancefield*.

solid impurities collect there, and sink down to the bottom, or apex of the cone, whence they are from time to time blown off, being first stirred up, if necessary.

IX. The *steam-pipe* conveys the steam from the boiler to the engine. As to its dimensions and resistance, see Article 52 of this Division. Besides the throttle-valve or regulator, by which the supply of steam to the engine is controlled, the steam-pipe of every boiler should be provided with a perfectly steam-tight *stop-valve*, to be shut when the boiler is not in use.

X. *Safety-valves*, for letting the steam escape from the boiler when its pressure tends to rise too high, will be further considered in a subsequent Article. Every boiler should have two; and of these one is usually placed beyond the control of the engineer.

XI. The *vacuum-valve* is a safety-valve opening inwards, to admit air into the boiler, and so to prevent it from collapsing, in the event of the steam within it falling below the atmospheric pressure.

XII. The *pressure-gauge* shows to the engineer the excess of the pressure within the boiler above that of the atmosphere.

XIII. The *water-gauge* shows to the engineer the level of the water in the boiler; and especially, whether it stands high enough to cover all those parts of the boiler which are directly exposed to the fire. The old form of water-gauge consists of three cocks at different levels; one at the proper level of the water, another a few inches above that level, and a third a few inches below. By opening these the engineer can ascertain the level of the water approximately. The new form consists of a strong vertical glass tube, communicating with the boiler above and below the proper water-level through cocks, which can be shut if the tube is accidentally broken. The level of the water is visible in this tube. Every boiler ought to be provided with *both* forms of water-gauge, the cocks and the glass tube, so that if the tube should be choked or broken, the cocks may be employed.

XIV. A *steam-whistle* may be used, as in locomotives, to make signals; and it may also be acted upon by a pressure-gauge, or by a float, so as to give warning of the pressure rising too high, or the water-level falling too low.

XV. *Stays* are bars, rods, bolts, and gussets for strengthening the boiler, which will be further referred to in a subsequent Article.

XVI. *Clothing* for the outer surface of a boiler, to prevent waste of heat, is usually made of a layer of coarse felt, covered with a layer of sheet-iron, or of thin wooden boards. It is applied to steam-chests and other parts of boilers which rise above the weather-deck.

The principal parts and appendages of engines and boilers having been enumerated and described generally, those which require it will now be treated of in a more detailed manner.

67. *Grate*.—As to the area of fire-grate, see Article 64.

The *length* of a grate should not much exceed 6 or 8 feet, in order that the fireman may easily throw coals to the back of it. It may be as much *less* than 6 feet as the dimensions and figure of the boiler require. The *breadths* of grates range from about 15 inches to 4 feet; the most convenient breadths for firing being from 18 inches to 3 feet, or thereabouts.

To facilitate the even spreading of the fuel, the surface of the grate is in general made to *slope downwards* from the fur-

nace-mouth to the bridge at the rate of about *one in six*. Its clear height above the floor of the ash-pit should be at least $2\frac{1}{4}$ feet in front.

A grate consists of *fire-bars*, and of *cross bearers*, by which the bars are supported. The fire-bars are made in lengths of from 2 to 4 feet. They are from $\frac{5}{8}$ to $\frac{3}{4}$ inch broad on the top, and are often made to diminish to about half that thickness at the lower edge, in order to admit of the free entrance of air and escape of ashes. Their ordinary depth is about 3 inches. The breadth of the clear space between two bars is from one-half to two-thirds of the greatest breadth of a bar. At each side of each end of a bar there are *snugs* or projections, by which the breadth of the bar at its ends is increased, so as to be equal to the distance from centre to centre of the bars. When the bars are laid upon the cross bearers, with the snugs touching each other, the proper spaces are left between their intermediate parts.

The clear height of the "*crown*," or roof of the furnace above the grate bars, is seldom less than about 18 inches, and often considerably more. As the crown of the furnace forms part of the heating surface of the boiler, a greater height is desirable in every case in which it can be obtained; for the temperature of the boiler plates being much lower than that of the flame, tends to check the combustion of the inflammable gases which rise from the fuel. As a general principle, a *high furnace is favourable to complete combustion*.

The height of the furnace is limited in practice, sometimes by the necessity for having flues or tubes traversing the water above it; and always by the necessity for having a sufficient depth of water above the highest part of the heating surface: that is to say, about 12 or 15 inches.

The *quantity of air* required for complete combustion of the best qualities of coal is thus made up:—For the supply of the oxygen that combines with the fuel, 12 lbs. per pound of fuel burned; for the dilution of the products of combustion, from 6 lbs. to 12 lbs. more; making, in all, from 18 to 24 lbs. of air per pound of fuel. The volume of 1 lb. of air at ordinary atmospheric temperatures is about 13 cubic feet; that volume is expanded from sixfold to sevenfold by the heat of the furnace, and contracts again by the time it reaches the uptake, to from twice to four times the original volume.

The best mode of insuring the complete combustion of the volatile parts of the coal consists in admitting air *above* the fuel to burn the gas, and *below* it to burn the coke. This requires the use of a deep fire, with openings and valves for air in the door, or in the furnace-front, and with a door to regulate the admission of the air to the ash-pit. The fire-door should be double, with an air-space between its two plates, and the apertures so arranged that heat cannot be radiated directly through them: thus much of the heat which would otherwise be lost by radiation through the fire-door is saved by being communicated to the air that enters through it.

68. *Strength and Construction of Boilers*.—The only figures for the *shells* of boilers which are safe against bursting by internal pressure, without the aid of stays, are the cylinder and the sphere, as to which see the Third Division, Article 30.

Portions of boiler-shells which are flat, or which otherwise deviate from the cylindrical and spherical figures, are strengthened by means of *stays*. The usual *pitch* or distance apart of the

stays of marine boilers is from 12 to 18 inches. The iron of the stays ought not to be exposed to a greater working tension than 3000 lbs. on the square inch, in order to provide against their being weakened by corrosion. This amounts to making the *factor of safety* of the stays for the working pressure from 16 to 20, nearly. For cylindrical boiler-shells, the factor of safety is from 6 to 8; the working tension being limited to from 4500 to 6000 lbs. on the square inch.

Tubes for the passage of flame and hot gas are made of brass or of iron, and are from 2 to 4 inches in diameter for marine boilers. They are fixed tight in the holes in the tube-plates, either by driving ferules into their ends, or by rivetting up the edges of the ends themselves, so as to make them fit countersunk grooves which surround the holes on the outside of each tube-plate.

Cylindrical internal flues tend to collapse by the pressure of the steam. The intensity of the pressure required to make a flue collapse is found as follows: *Multiply 9,600,000 lbs. on the square inch by the square of the thickness, and divide by the length and diameter, all in inches.* When the flue is stiffened by rings outside, the length is to be measured from ring to ring.

The flat ends of cylindrical shells are sometimes connected with the barrels and flues by means of rings of angle iron; but such rings are liable to split at the angle; and therefore it is preferable to make the connection by bending the edges of the endmost plates of the barrel and flues.

The shells of marine boilers are usually double-rivetted at the bottom, and single-rivetted at the sides and top. Horizontal overlapped joints should have the overlapping edges facing upwards on the side next the water, that they may not intercept bubbles of steam on their way upwards. The joints in horizontal flues should be so placed that the overlapping edges shall not oppose the current of gas.

Those parts of boilers which are exposed to more severe or more irregular strains than the rest, or to a more intense heat, should be made of the finest iron, such as Bowling or Lowmoor. This applies to the sides and crowns of internal furnaces, to tube-plates, to bent plates at the ends of cylindrical shells, &c.

69. *Testing or Proving Boilers.*—Before any boiler is used, its strength ought to be tested by means of the pressure of water. The water is usually forced in by pumps. The *testing pressure* should be *not less than double the working pressure, and not more than half the bursting pressure*; that is to say, as the bursting pressure should be six times the working pressure, the testing pressure should be between twice and three times the working pressure. About *two and a half times* the working pressure is a good medium.

Another method of producing pressure for the testing of boilers without danger, in the absence of suitable pumps, is that contrived by Dr. Joule. It consists in filling the boiler quite full of water, so that there is no room for steam, and then heating it, until the tendency of the water to expand produces the required pressure.

In everything that relates to the strength and testing of boilers, the *pressure* is to be understood to mean the *excess of the pressure within the boiler* above the atmospheric pressure.

The pressure of water is to be used in testing boilers, because of the absence of danger in the event of the boiler giving way to it. Testing boilers by steam-pressure is very dangerous.

70. *Safety-valves.*—One at least of the safety valves of a boiler should be loaded directly, and not through the medium of a lever.

The load, whether applied through a lever or to the valve directly, may be produced either by a weight, or by the elastic force of a spring.

The seat of a safety-valve should be accurately turned and ground to the form of a spherical belt, in order that the valve may fit it in different positions. The following rule for the area of the narrowest part of the outlet of a safety-valve agrees well with good practice: *Multiply the greatest weight of water to be actually evaporated in lbs. per hour by 0.006; the product will be the required area in square inches.*

71. *Feed and Blow-off Apparatus.*—There should be duplicate sets of feed-pumps, so that if one set breaks down the other may be used. As to the proper supply of feed-water, see Article 45 of this Division. The feed-pumps are worked by the engine itself when it is in motion; but when it is standing still, and it becomes necessary to feed the boiler, they are driven either by hand, or by a donkey-engine. For all marine boilers of considerable size, a donkey-engine is necessary; and it may be used not only to feed the boiler, but to drive the starting and reversing gear of the valves when required, and perform other miscellaneous duties.

When a boiler is fed with salt-water, the brine is either *blown-off* periodically, or extracted by means of a *brine-pump*. To find the volume of brine to be discharged in a given time, divide the *gross* supply of feed-water by the proportion in which the brine is to be stronger than sea-water, say about 2. (See Article 45 of this Division.)

The brine is discharged at a temperature on an average 140° or 150° Fahr. higher than that at which the feed-water is drawn from the hot-well. In order that the apparatus of tubes and casing already mentioned in Article 66 of this Division may act with the greatest possible efficiency in transferring heat from the hot brine to the feed-water, the surface of the tubes should amount to about $\frac{1}{16}$ th of a square foot per pound of brine discharged per hour; or $6\frac{1}{2}$ square feet per cubic foot of brine discharged per hour.

It may, however, be sometimes difficult or inconvenient in practice to obtain so large a surface.

The *Injector* drives the feed-water into the boiler by the impulse of a jet of steam, whose particles condense at the instant of striking the stream of cold water which they impel. This instrument, though not economical of heat, is convenient because of its simplicity, and its needing no engine to work it. It is not much used for marine boilers; and it would probably not be advisable to use it to the exclusion of the feed-pump; but still it may be an useful appendage to a boiler, for the purpose of supplying water when the feed-pumps are not working.

To estimate the *sectional area of passage* required at the narrowest part of an injector, the following approximate rule may be used: *Divide the volume of water to be supplied in cubic feet per hour by 800, and by the square root of the pressure of the steam in atmospheres; the quotient will be the required area in square inches.* For circular inches, use 630 as a divisor instead of 800; the square root of the area in circular inches will be the diameter in inches.

SUPPLEMENT TO THE SIXTH DIVISION.

I. ADDENDUM TO ARTICLE 9, PAGES 250, 251:—

Deflecting Fixed Blades and Deflecting Rudders.—Mr. Rigg has proposed to apply his fixed deflecting blades to paddles as well as to the screw. For further information respecting their application to the screw, under the name of the Duplex Screw Propeller, see the Transactions of the Institution of Engineers in Scotland and Scottish Shipbuilders' Association for the 20th December, 1865.

Considering that fixed deflecting blades are applicable only to vessels which have an after-sternpost abaft the screw, and not to those which have a balanced rudder and no after-sternpost, Mr. James R. Napier and the Editor of this Treatise have proposed to make the balanced rudder itself deflect the water gradually into a right aft direction by so shaping it that when the helm is amidships the forward edge of the rudder shall stand tangentially to the current coming from the screw, and that its after edge shall point right aft.

Fig. 1.



Fig. 1 shows the construction for finding the proper inclination of the surface of the deflecting rudder at its forward edge to a fore-and-aft line.

Draw CA to represent the speed of advance of the screw through a solid (= pitch \times turns per second); cut off from it CB to represent the *real slip* of the screw relatively to the water of which it lays hold; so that BA shall represent the speed of the vessel relatively to the water at her stern *before* that water is acted on by the screw, and AB the backward speed of that water relatively to the vessel.

On BC as a diameter draw a circle. Lay off the angle ACD to represent the obliquity of any given strip of the surface of the screw to a fore-and-aft line, and let CD cut the circle in D. Join BD; this will represent in direction and magnitude the velocity which the strip of screw-surface in question impresses on the particles of water. Join AD; this

will represent the resultant velocity and direction of the motion, relatively to the vessel, of the particles of water after having been acted upon by the given strip of screw; and this also will be the proper direction for a tangent to the surface of the rudder at that part of its forward edge which receives the water from the given strip of screw-surface.

By repeating the same process for a series of strips of the screw-blade at different distances from its axis, a sufficient number of tangents may be determined to enable a mould for the rudder to be constructed. (See Transactions of the Institution of Naval Architects, 1866.)

II. ADDENDUM ON STEAM-SHIP CAPABILITY.

In this addendum algebraical symbols are employed, as without their aid both the investigation and its results would be excessively voluminous.

Let D be the displacement in tons of a steam-ship at mid-passage (that is, when half her stock of fuel has been burnt);

n D (= from 0.33 to 0.4 D) the weight of her hull and equipments;

V, her intended mean speed, in knots (in what follows it is assumed that the mean speed and the speed at mid-passage are equal: this is approximately but not exactly true);

T, the length of her intended trip, with one supply of coals, in nautical miles;

L, her cargo, or profitable lading of any kind;

Then the weight of her engines and boilers in tons is (see First Division, Article 99, pages 39, 40);

$$E V^2 D^{\frac{1}{3}} \quad (E \text{ ranging from } 0.001 \text{ to } 0.0016);$$

and the weight of her stock of fuel,

$$F V^2 D^{\frac{1}{3}} T \quad (F \text{ ranging from } 0.000006 \text{ to } 0.000002);$$

so that the displacement available for profitable lading is

$$L = (1 - n) D - (E V^2 + \frac{1}{2} F V^2 T) D^{\frac{1}{3}} \quad (1)$$

The above equation, when the coefficients n , E , and F have been determined, enables the following problems to be solved.

(1.) Given, the mid-passage displacement D, speed V, and length of trip T; to find the cargo L: calculate the value of the right-hand side of the equation as it stands.

(2.) To find the displacement of the smallest steam-vessel that will run a given trip at a given speed without cargo: make L = 0; then

$$D = \left(\frac{E V^2 + \frac{1}{2} F V^2 T}{1 - n} \right)^3$$

(3.) To find the displacement of the smallest steam-vessel that will run a given trip at a given speed with a given cargo; solve the equation (1) as a cubic equation, with $D^{\frac{1}{3}}$ as the unknown quantity.

(4.) To find the greatest speed at which a vessel of a given displacement with a given cargo can run a given trip without coaling. Solve equation (1) as a cubic equation, with V for the unknown quantity.

(5.) Suppose the condition to be laid down, that the cargo shall be a certain fraction of the mid-passage displacement, viz., $L = mD$; then we have the following expression for the displacement of the smallest vessel with a given fraction of her mid-passage displacement for cargo, that can make a given trip at a given speed without coaling;

$$D = \left(\frac{E V^2 + \frac{1}{2} F V^2 T}{1 - n - m} \right)^3 \quad (4)$$

EXAMPLE OF THE LAST PROBLEM.

Suppose $n = 0.36$; $m = 0.2$; $E = 0.0012$; $F = 0.00001$; $T = 24,000$ nautical miles; $V = 10$ knots; then $D = 30^3 = 27,000$ tons.

The preceding formulæ relate to mechanical questions alone. For examples of the application of the same principles to commercial questions, see Mr. Charles Atherton's work on "Steam-ship Capability," also a paper by Mr. James R. Napier "On the most Economical Speed for a fully laden Cargo-steamer" in the Proceedings of the Philosophical Society of Glasgow, November and December, 1865.

III. ADDENDUM AS TO THE MOST ECONOMICAL RATE OF EXPANSION.

By the aid of the right-hand diagram in Plate $\frac{f}{f}$, the following question can be solved: To find the most economical rate of expansion in a given engine.

The data required are—the proportion of back pressure to absolute initial pressure of the steam in the cylinder; and the proportion of cost of engine to cost of steam.

Under cost of engine for a given period are to be included interest of the price of the engine (exclusive of boilers), depreciation, repairs, engineers' pay, and other working expenses depending on the dimensions of the engine; also the value of the stowage-room occupied by it in the ship.

Under cost of steam are to be included interest of the price of the boilers, together with their depreciation and repairs, cost of fuel, firemen's wages, and other working expenses depending on the expenditure of steam; also the value of the stowage-room occupied by the boilers and fuel in the ship.

Let A C d b in Fig. 2 be conceived to represent the right-hand diagram in Plate $\frac{f}{f}$; A B = 2 A b being the whole stroke, A C the absolute initial pressure, and A D the curve, whose horizontal abscissæ represent effective cut-off, and its vertical ordinates mean absolute pressures. Lay off A E to represent the back pressure, and draw E F parallel to A B.

If the cost of steam has been estimated for a given rate of expansion, multiply it by that rate to find the cost of full steam.

Produce F E to G, making

$$E G = A B \frac{\text{cost of engine}}{\text{cost of full steam}}$$

From G draw G H touching the curve A D; and from the point of contact H let fall the perpendicular H K on A B, cutting E F in L. Then K will represent the most economical effective cut-off, $\frac{A B}{A K}$ the corresponding rate of expansion, H K and H L the corresponding mean absolute and mean effective pressures, and $\frac{E L}{E G}$ the ratio of cost of steam to cost of engine, when working with the greatest economy. (See paper by the Editor of this Treatise in the Transactions of the Institution of Naval Architects for 1866; and the Transactions of the Royal Society of Edinburgh for 1851.)

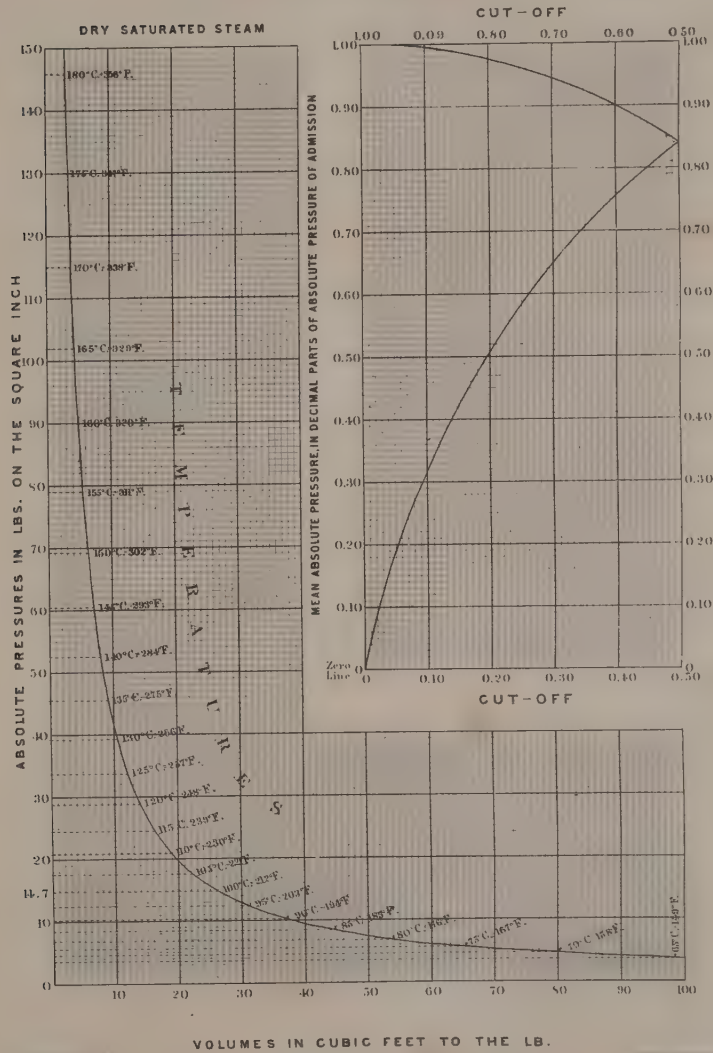
IV. REFERENCE TO RIGHT-HAND FIGURE IN PLATE $\frac{f}{f}$.

This Figure shows a horizontal section, through the cylinders, of the pair of double-cylindred screw engines of which a thwartship section is given in the same Plate.

- | | |
|----------------------------------|--|
| A, A, high-pressure cylinders. | N, wheel-casing, containing wheel on |
| B, B, low-pressure cylinders. | engine-shaft in inside gearing with |
| C, exhaust-pipe. | pinion on propeller-shaft; multiple |
| D, condenser (surface). | of gearing, 2 $\frac{1}{2}$. |
| K, K, low-pressure slide-valves. | P, propeller-shaft. |
| L, L, high-pressure do. | Q, cold-water circulating pump, worked |
| M, reversing-gear and donkey. | by a crank on after end of engine-shaft. |

MECHANICAL PROPERTIES OF STEAM

by Col. J. Macquorn Rankine, C.E., F.R.S.



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DIVISION SEVENTH.

SHIPBUILDING FOR PURPOSES OF WAR.

ARTICLE 1.—*General Remarks.*—In all that relates to buoyancy, stability, strength, resistance, speed, propulsion, and handiness, the principles of shipbuilding for purposes of war are the same with those of shipbuilding for other purposes. The differences in detail, as regards the application of those principles, by which war-ships are distinguished from merchant ships, have already to a great extent been stated in the previous Divisions of this Treatise; but it may be useful here to give the following brief summary of them.

I. Owing to the greater proportionate strength of a war-ship, the *weight of her hull* usually is a larger fraction of her whole displacement than that of a trading ship; say, from 0·4 to 0·5, instead of from 0·3 to 0·4, and that *exclusive* of armour.

II. The *decks*, and the beams, stanchions, and bulkheads which support them, must have strength sufficient to bear the weight of the guns. The *weight of a gun* ranges, in smooth-bored guns, from 500 to 700 times, and in rifled guns, from 600 to 800 times, the weight of the *service charge of powder*.* The weight of the carriage is about one-fifth of that of the gun.

III. In ships whose sides are pierced with *port-holes* for guns, the longitudinal strength thus taken away must be supplied by adding sufficiently to the sectional area of the skin and framing above and below the ports. In iron ships of war the skin above and below the tiers of port-holes is doubled, and longitudinal stringers are fixed outside. Port-holes were formerly made 4 feet square, or nearly so; at present, though the height is not much diminished, the breadth is made only 2 feet, or thereabouts, and in some cases just wide enough for the muzzle of the gun. Their distance apart from centre to centre ranges from 10 to 16 feet. The lowest port-sills should be at such an elevation above the load-water-line, that they may not be immersed by a heel of from 10° to 12° in two-decked or three-decked vessels, or from 15° to 20° in single-decked vessels (that is, vessels with but one tier of guns). In ships of war as built without armour, each port-hole is closed either by a *port-lid* opening upwards, or by a pair of port-lids opening one upwards and the other downwards.

IV. As to *strength*, general and local, the same methods of

construction that are advantageous in merchant ships are also advantageous in ships of war, and may be carried out with less regard to expense; and in particular, longitudinal framing, water-tight transverse bulkheads, and a double bottom; the space between the outer and inner skins of the double bottom being divided into water-tight cells by the webs of the longitudinal ribs or keelsons. The depth of that space ranges from 4 to 3 feet; the width of the cells is usually about 6 feet. In iron ships of war, it is advisable to have the upper deck, and sometimes the main deck and lower deck also, completely plated with iron or steel; and the usual position for such plating is between the beams and the planking. Platform decks in such vessels are made with iron plates, without planking.

V. *Stability* requires to be more carefully studied in ships of war, than in merchant ships, because of the heavy weights to be carried above the water-line in the shape of guns and armour, and the importance of easy motion and dryness to the safety and efficiency of the ship and her armament. In particular, *easy rolling* is to be as far as possible combined with sufficient *stiffness* laterally, and *liveliness* in pitching and scending. The means of obtaining those qualities have been fully explained in the First Division.

VI. *Handiness* also is of special importance in ships of war. Hence, for the sake at once of handiness, stability, and liveliness, any excess of length above that which is necessary to the attainment of the required speed with economy of power (see page 82) is to be avoided. In ships designed by M. Dupuy de Lôme, moderate length above water is combined with fine entrance lines, by means of a projecting beak below water. Balanced rudders and the twin-screws are favourable to quickness in manœuvring.

VII. *Sea-going War-steamers* should be designed with a view to being weatherly, swift, and handy under sail alone, as well as under steam, in order to save their stock of coal for battle and chase. It is essential that the engines should be provided with a worm wheel (V, Plate $\frac{1}{2}$) to turn them by hand from time to time, while the ship is under sail, and so prevent the bearings from sticking. The screw is often made to lift out of the water when not in use. The preferable forms of engine and boiler are those which can be contained wholly below the load-water-line, and, if possible, below also that part of the ships' side which is exposed by a roll of 10° or 12°; hence the cylinders are usually horizontal, except in sharp-floored vessels, which admit of their being inclined.

* Guns are denominated from the weight of the spherical cast-iron shot which they are capable of taking in, being usually a little more than 4 times the weight of the service charge of powder. The weight of a cast-iron sphere in pounds = cube of diameter in inches \times 0·134 nearly; weight of a steel sphere in pounds = cube of diameter in inches \times 1·48 nearly. Cylindrical shot for rifled-guns are usually about double the weight of spherical shot of the same diameter, and about 8 times the weight of the service charge of gunpowder with which they are fired; but the gun still generally takes its name from the weight of the spherical cast-iron shot that fits its bore.

VIII. The principles of *steam-ship capability* (explained in Addendum II. to the Sixth Division, page 292) may be applied to war-steamers, by substituting in the equations the weight of guns, stores, and armour, for that of cargo.

There remains to be considered in the subsequent Articles of this Division, the subject of armour for war-ships, depending on the explosive energy of gunpowder, the destructive power of projectiles, and the resistance of armour-plates and backing.

2. The *Explosive Energy of Gunpowder*, completely burned, is from 240,000 to 300,000 foot-lbs. per pound;^a but owing to incomplete combustion, recoil, and other causes of loss, the energy actually communicated to cannon shot ranges between 144,000 and 192,000 foot-lbs. per pound, according to the quality of the powder.

The weight of the projectile ranges from four times to eight times that of the powder; the former ratio being usual in spherical shot, the latter in cylindrical. The velocity with which the shot leaves the gun may be calculated thus: divide the energy due to the powder by the weight of the shot; the quotient is a height in feet, which is to be multiplied by 64.4; the square root of the product will be the initial velocity of the shot in feet per second.

During the flight of the shot, part of its energy is lost through the resistance of the air, which diminishes its speed: the energy which remains in the shot diminishes proportionally to the square of the speed. But in designing armour-plated vessels, such loss of energy should not be reckoned upon; for their armour may have to resist shot fired at close quarters.

When the shot strikes the armour-plate, a further waste of energy takes place through the disfigurement of the shot itself. It has been shown by Sir W. G. Armstrong, and corroborated by the experiments of the Iron Plate Committee, as quoted by Mr. Fairbairn, that when the shot is of cast-iron, it flies to pieces on striking the armour-plate; and that in producing that effect about half its energy is wasted. If of wrought iron or soft steel it is compressed and flattened, and about a fifth of its energy is wasted. If of hardened steel it is but slightly compressed; and if broken, it splits longitudinally, and less than a tenth of its energy is wasted. In designing ships' armour, provision must be made for the case of the enemy using the most efficient material for projectiles; hence it is safest to estimate the energy of the blow at from 144,000 to 192,000 foot-lbs. per pound of powder, without deduction.

3. *Action of Shot on Armour-Plate.*—The first effect of a shot striking an armour-plate is to produce an indentation, in the neighbourhood of which the iron is compressed. The compression is transmitted both forwards and sideways, diminishing in intensity as it is spread over layers of iron of gradually increasing area, and accompanied by tension in a transverse direction. The energy of the shot is expended in producing that strained condition of the iron. The *dynamic resistance*, or power of the plate to resist the blow, is equal to the quantity of mechanical work required to bring the plate into the most highly strained condition which it can bear without rupture. Should that quantity of work be greater than the energy of the shot, the

plate withstands the blow; should it be less, the plate gives way, either by the formation of a number of radiating cracks, spreading from a point opposite the shot on the inner side of the plate, or by the punching out of a piece of a roughly conical figure, spreading from the indentation at the outer side, or by making a hole surrounded by a burr. The first mode of giving way, by cracking or bursting, is held to be a sign of an inferior quality of materials or of workmanship; the second, by punching, of a better quality; the last, by a burred hole, of the toughest and best.

Besides the quality of the iron, the dynamic resistance of a plate of a given thickness depends on the volume of metal put into a strained condition by the blow of the shot, and on the distribution of the stress in that metal. Owing to the imperfect state of our experimental knowledge, there does not yet exist any complete and exact theory of the laws of the dynamic resistance of plates to shot; but from such investigations as are possible in the present state of knowledge, it is clear, that for a plate of a given thickness *there is a certain diameter of indentation for which the dynamic resistance is a minimum*; becoming greater for a larger indentation, because of the increased volume of metal throughout which the stress acts; and greater also for a smaller indentation, because of the way in which the stress is distributed.

This may be understood independently of mathematical investigation, if it is considered that a shot impelled by a charge of powder otherwise sufficient, may fail to burst through a plate either through being too large or too small; if too large, by its impact being spread over too great an area; if too small, by the energy being spent in burying the shot amongst the particles at the outside of the plate; so that for a given thickness of plate, there is a certain diameter of shot which will pass through the plate with a *minimum charge of powder*, or in other words, which reduces the dynamic resistance of the plate to a minimum.

The results of experiment, so far as they have yet gone, appear to confirm this view, and to show that the most efficient diameter of shot must be somewhere about *once-and-a-half* the thickness of the plate. They further show (as Mr. Fairbairn has pointed out in his work on Iron Shipbuilding) that with shot of the proper diameter, *the dynamic resistance of a plate varies nearly as the square of the thickness multiplied by the diameter of the shot*; and as the most efficient diameter of shot is proportional to the thickness of the plate, it follows that *the minimum dynamic resistance varies nearly as the cube of the thickness of the plate*.

That resistance may be calculated approximately by the following RULE: (I.) *Multiply the cube of the thickness in inches by 16,000; the product will be the minimum dynamic resistance in foot-lbs., nearly.*

When this rule is compared with the rule given in the preceding Article for the energy of hardened steel shot at short ranges, the following rules are deduced from it:—

II. For a rough approximation to the greatest charge in pounds of powder which a given armour-plate will just withstand, when fired with hardened steel shot of the most efficient diameter: take, for weak powder, *one-ninth*, or for strong powder, *one-twelfth*, of the cube of the thickness in inches; and conversely.

^a Gun Cotton is considered to have about three times the explosive energy of the same weight of gunpowder.

III. For the thickness in inches: *extract the cube root of 9 times the charge in lbs., for weak powder, or 12 times the charge in lbs., for strong powder.*

With *cast-iron shot*, about double the weight of powder is required to produce the same destructive effect on armour-plates.*

4. *Quality of Iron in Armour-plates.*—The iron of armour-plates should be tough and homogeneous, and free from all visible traces of the layers of which it is made up; and, according to Mr. Fairbairn's experiments, should have the following strength against different modes of breaking:—

TENACITY; average about 22 tons per square inch; should not be less than 20: ultimate elongation of bar when torn about 0.2 of original length.

CRUSHING LOAD of a small block; average about $91\frac{1}{2}$ tons per square inch; should not be less than 90: ultimate compression, about *half* of original length.

5. *Backing of Armour-plates.*—Backing composed wholly of timber (as when armour-plates are simply bolted upon a wooden ship) adds little to the power of an armour-plate to resist penetration, its principal use being to stop shot and shell after they have passed through the armour-plate, and to diminish the vibration communicated to the hull of the ship by a blow; and for that purpose it should be made as thick as possible, by filling in the spaces between the frames with solid wood.

It is otherwise when the wooden backing has a rigidly framed iron skin behind it (see Plate $\frac{B}{3}$); for then the compression of the wood between the armour-plate and the iron skin takes up a considerable part of the energy of the blow; so that the backing not only serves to stop shot that may penetrate through the armour-plate, and to deaden vibration, but adds to the dynamic resistance that must be overcome before the plate can be penetrated.

In actual examples, the skin-plates are about one-eighth of the thickness of the armour-plates, sometimes single and sometimes double, and are supported and stiffened inside by vertical frames 10 inches deep, or thereabouts, spaced to about 2 feet; and outside by longitudinal frames or stringers of about the same depth, from 3 to 6 feet apart in the earlier examples (Plate $\frac{B}{3}$), and about 2 feet apart in the later examples. Between and outside those stringers is fixed the wooden backing of teak or oak, from twice to four times the thickness of the armour-plates. If twice the thickness, it is in one layer laid longitudinally; if four times, in two layers, the inner sometimes vertical, the outer longitudinal, as in Plates $\frac{B}{3}$, $\frac{B}{4}$. The armour-plates are fastened on with through-bolts, about 2 feet apart, of a diameter equal to from one-half the thickness of the thinner kinds of armour-plates to one-third of the thickness of the thicker kinds. The heads of the bolts are outside, and are countersunk in conical holes: the nuts, which are usually double, are inside, and have sometimes thick india-rubber washers between them and the skin plates.

The exact extent to which timber backing over an iron skin adds to the dynamic resistance is still very uncertain; but the following estimates may be taken as approximately true:—

I. An armour-plate backed with *four times* its thickness of teak or oak over an iron skin is nearly equivalent in dynamic

resistance to an unbacked plate of about *once-and-a-third* its thickness.

II. An armour-plate backed with *twice* its thickness of teak or oak over an iron skin is nearly equivalent in dynamic resistance to an unbacked plate of about 1.055 times its thickness.†

Compound Backing‡ consists of alternate layers of timber and plate-iron, in the proportion of $4\frac{1}{2}$ inches of timber to $\frac{1}{2}$ inch of iron, or nearly so, set with their edges outwards, and fastened together with bolts. It is placed between the outer armour-plate and an inner armour-plate of about one-fourth of the thickness of the outer armour-plate. Between the inner armour-plate and the skin of the ship is a second backing of wood alone; the whole bolted to the ship with through bolts, and the ship's skin being stiffened as already mentioned with longitudinal outside stringers. The use of the second backing is mainly to deaden vibration; that of the inner armour-plate partly to stop shot which may pass through the outer armour-plate, and partly to add to the dynamic resistance; while the first or compound backing of alternate layers of wood and iron serves to add to the dynamic resistance in a greater proportion than is possible with wood alone. The following may be taken as a rough estimate of the comparative dynamic resistance of this combination of armour:—

III. An armour-plate backed with three times its thickness of compound backing (including an inner armour-plate of about one-fourth of the thickness of the outer), and with a second backing of wood of about half the thickness of the first backing, over a properly framed iron skin, is equivalent to an unbacked plate of about 1.6 times the thickness of the outer plate.

Paper has been proposed as a material for backing; but although there can be no doubt of its great strength, if properly prepared, data are still wanting to enable its resistance to be estimated.

In some experiments, a *facing* of wood,§ outside an armour-plate, has been found to increase materially the resistance to the penetration of shot.

6. *Various positions of Ships' Armour.*—The armour of ships may be classed under three heads, according to its position and purpose—viz., I. Armour for the protection of the ship as a whole; II. Armour for the protection of particular parts of the ship, such as magazines, and engine and boiler rooms; III. Armour for the protection of the guns and crew.

I. Armour for the protection of the ship as a whole consists essentially of a belt, and an iron or steel-plated deck; the belt running round the ship between wind and water; the depth to which the belt extends below water being that corresponding, at the extreme breadth of the ship, to a heel of from 10° to 12° , and above water, about one-third or one-half of the depth below water as a minimum, and as much additional depth as the constructor may think advisable; the iron or steel-plated deck, with plates of from half an inch to two inches thick, not being above the level of the highest part of the side armour. In the

* The following is the formula deduced from a provisional mathematical theory: let d be the diameter of the shot; t the thickness of the plate; c and n constants; then dynamical resistance nearly =

$$c(n d + t)^3 \cdot \frac{t}{n d}$$

Value of n , probably about $\frac{1}{2}$; of c , 2870 foot-lbs., nearly, for dimensions in inches.

† Provisional formula for dynamic resistance of a backed armour-plate. Let diameter of shot = d ; thickness of armour-plate = t ; thickness of backing = b ; n , c , constants; then dynamic resistance = nearly—

$$c(n d + t)^3 \cdot \frac{t}{n d} + c'(n d + t + b)^3 \cdot \frac{b}{n d + t}$$

n probably = about $\frac{1}{2}$; c = 2870 foot-lbs. for dimensions in inches; c' , for teak and oak backing, 47; for compound backing, $\frac{1}{10}$ iron, $\frac{1}{10}$ wood (b measured to the inner side of the inner armour-plate), c' = 280.

‡ Mr. James Chalmers' system.

§ Proposed by Commander R. A. E. Scott, R.N., and Mr. Samuda.

American "Monitors," the deck, which is only a foot or sixteen inches above the water, is protected by plates two inches thick, with a solid timber backing below them. At the bow, the armour-belt, built up solid inside with wood, may be formed into a beak or ram if required. A long fine bow is sometimes left unprotected for the sake of lightness and liveliness, and divided into water-tight cells; immediately abaft of which is a shot-proof bulkhead. At the stern, it is desirable that the immersed and armour-belted part of the ship should overhang the screw and the rudder. If that overhanging part is bluff and rounded, it adds considerably to the resistance of the vessel; but this may be avoided by giving it fine water-lines, so as to form an after-beak.

II. Armour specially for the protection of magazines, engines, and boilers, consists in general of armour-plated transverse bulkheads, or of shell-proof decks and platforms. The coal-bunkers are often so arranged as to give additional protection to engines and boilers against shot or shells which may penetrate through the side armour of the ship.

III. Armour for the protection of the guns and crew may be much varied in its extent and arrangement.*

The simplest arrangement consists of a shot-proof bulkhead or shield near the bows of the vessel, with portholes for one or two bow-chasers or guns to be fired right ahead; and this is suitable for the "end-on" system, in which the guns are trained by the steering of the vessel. Next in order is the arrangement which has two such shields: one near the head, the other near the stern.

For a still greater armament, some vessels are built with a central battery, or "fighting-box," in the middle of their length, of a nearly rectangular shape, inclosed with armour suited to resist the charges of powder to which it is to be exposed. Such a central battery is exemplified in Plates $\frac{B}{5}$, $\frac{B}{6}$, $\frac{B}{7}$. Its length depends on the number of protected guns on one side of one deck, and on the distance from centre to centre between the ports.

In more recent examples, the sides of the unprotected part of the ship form a triangular recess at each corner of the central battery, so as to give room for four ports in its forward and after sides, through which guns can be fired in directions deviating a few degrees only from a fore-and-aft line. Another way of effecting the same object is to have guns pointing fore and aft in a part of the central battery that rises above the weather decks of the unprotected ends of the vessel.

The weight of armour required for a central battery has been estimated at about 30 tons per gun for a single gun deck, and 20 tons per gun for two decks.

For the fullest armament possible on the broadside principle, the ship is armour-plated up to the upper deck from stem to stern. The upper deck is either without guns, or has a few guns only, for firing ahead and astern, sometimes protected by transverse armour bulkheads or shields.

The armour plates are usually doubled round the port-holes, because of the weakness or absence of backing there.

Turret-ships may be considered to possess, on the whole, the best arrangement for carrying an armament consisting of from two to four very large guns. They may be divided into two classes: the British Turret-ships of Captain Cowper P. Coles, R.N. (see

Plate S), and the American "Monitors" of Captain Ericsson. In both kinds of turret-ship, one or more cylindrical shot-proof turrets rise from a deck a little way above the load-water-line, and suitably supported from below by beams, bulkheads, and stanchions. Each turret is usually about 24 feet in diameter inside, and carries sometimes one very large gun, but more commonly a pair of such guns, side by side, pointing the same way. The weight of the turret, together with its supports, is from four to six times that of the guns that it carries. It turns about a vertical axis; and its weight may be carried either by that axis, or by rollers resting on a circular turntable. It can be turned into any required position by a donkey engine. The roof commonly consists of a grating of strong iron bars (in some examples four inches deep by three inches broad), resting on iron beams, and covered with an iron or steel plate an inch thick or thereabouts, perforated with holes for ventilation. In the "Monitors," each port-hole is closed by a stopper, consisting of a large wrought iron crank, turning about a vertical axis. The "pilot-house," or "conning-house," is a small fixed turret, either separate from the gun turrets (as in Plate S), or standing on the top of a fixed wrought iron pillar in the axis of a gun turret. An American turret is about nine feet high; a British turret about fourteen feet, six feet of which rise above the upper deck.

In the British turret-ship, there is an upper or weather-deck, through a circular hole in which the turret rises, surrounded by a ring-shaped bevelled *glacis-plate* (as in Plate S). Round that deck are hinged bulwarks, which can be turned down in action. In the "Monitor" form, the deck on which the turret stands is itself the weather-deck. In both cases the exposed decks are plated with iron or steel from half-an-inch to two inches thick, under or over the timber planking. The armour of the sides comprises in both cases a belt of suitable depth between wind and water, extending up to the level of the deck that carries the turret; in the British form that belt is flush with the side; in the American form it overhangs. In the British form, the armour-plating alongside the turret or turrets is carried up to the level of the upper deck, and also round afore and abaft the turrets in the form of a pair of curved bulkheads. The armour of the turrets does not extend below the level of the weather-deck (see Plate S).

Owing to the fewness and small size of the openings in armour-plated ships, their proper ventilation requires special appliances, such as fans and air-ducts, already referred to in page 211, and illustrated in Plates $\frac{B}{5}$, $\frac{B}{6}$, $\frac{B}{7}$, $\frac{B}{8}$. In "Monitors," where the weather deck is almost at the water-line, the inlets for fresh air and outlets for foul air are vertical trunks, rising to such a height as to be safe from the entrance of water at their upper ends. In ships with hollow iron masts, those masts may be used as chimneys to carry off foul air.

DESCRIPTION OF PLATE S.†

Plate S is a design for a sea-going turret-vessel armed with four 600-pounder guns. The sides of the ship and walls enclosing the turret-battery are represented in defensive powers by a maximum weight per square foot of side of 467 lbs., minimum 433 lbs. The main deck is armour-plated with $1\frac{1}{2}$ inch plates,

* The classification in the text agrees nearly with that adopted by Mr. Scott Russell.

† Furnished by Mr. C. F. Honwood, N.A.

and the openings in the deck are capable of being closed against shot or shell.

The boats are stowed on a hurricane deck over the turrets, and the anchor fittings are specially arranged to suit the system.

The bulwarks, shown broken off in elevation, fig. 1, are made to turn down all round the vessel.

The guns of the turrets (as shown by the *arcs of training* marked on the plan) are capable of a concentrated fire of 2400 lbs. from either bow, as at *a, a, b, b*, fig. 2, and a direct fire in a line with the keel ahead, *c, c*, and astern, from two 600-pounders.

Fig. 3 is a cross-section at the turrets, showing space around turret for working the turret and serving ammunition.

Fig. 4 shows plan of turret at deck and at top, showing apertures for air, and look-out for the captain of the guns. The following are the principal dimensions—Length, 260 feet; beam, 54 feet; draught, 23 feet; height of deck above water, 10 feet 6 inches; height of port-sill, 12 feet 6 inches above water; horse-power, 800 nominal; tonnage, old measurement, 3530.

DESCRIPTION OF THE "HROLF KRAKE."

As an example of an actual turret-ship which has stood the test of battle, may be taken the *Hrolf Krake*, built for the Danish Government, from Captain Coles' design, by Messrs. R. Napier and Sons. She is fitted with two turrets, each of which are capable of carrying two 120-pounder guns, thus giving a broadside of four 120-pounders.

Her principal dimensions are:—Length, 183 feet 6 inches; beam, 38 feet 2 inches; draught of water, 10 feet 4 inches; armour-plating $4\frac{1}{2}$ inches thick; height of deck above water in midships, 4 feet 9 inches; height of port-sill above water, 6

feet 4 inches; horse-power, 235, nominal; speed, 10 knots; tonnage, 1246, old measurement.

DESCRIPTION OF THE TURRET-SHIP "HUASCAR,"

built for the Peruvian Government by Messrs. Laird Brothers.*

Armament:—one turret, forward, carrying two 300-pounder guns; broadside guns, two 40-pounders, rifled.

Length, 200 feet; beam, 35 feet; draught of water, 13 feet 3 inches; nominal horse-power, 300; speed, 11 knots; tonnage, 1166, old measurement.

Principal elements of two proposed sea-going cruisers of high speed, the general arrangements being similar to designs on Plate S.^o

	No. 1.	No. 2.
Length,.....	280 ft. 0 in.	315 ft. 0 in.
Beam,.....	50 " 0 "	52 ft. 0 "
Draught of water,.....	23 " 6 "	25 " 0 "
Height of deck above water,.....	11 " 0 "	12 " 0 "
Height of port-sill above water,.....	13 " 0 "	14 " 0 "
Horse-power, nominal,.....	800	1000
Tonnage, Old Measurement,.....	3325	4083
Offensive powers as represented by } weight of broadside in lbs. from } four 600-pounders,.....	2400 lbs.	2400 lbs.
Defensive powers as represented by } weight per square foot in lbs.,.....	Max. 467 lbs. Min. 433 "	Max. 507 lbs. Min. 473 "
Estimated speed in knots, average,.....	14 $\frac{1}{2}$	15
" " " on emergency,.....	16	17
Carries coals to steam a distance in } miles estimated at,.....	3300	3300
Thickness of armour,.....	6 ins.	7 ins.

These vessels are to have full sail-power, and the propellers are to be made to hoist, so as to insure the power of cruising under sail without steam, reserving the coals for emergencies.

* Furnished by Mr. C. F. Henwood, N.A. For further information on Turret-ships, reference may be made to the Report of the Committee on that subject, printed in March, 1866.

ADDENDUM TO THE THIRD DIVISION.

MR. BARNABY, in a paper read to the Institution of Naval Architects in 1866, proposed a method of increasing the power of pieces under tension, such as iron stringers and iron deck-plating, to resist sudden strains. It consists in reducing the quantity of material in the piece, so that its strength at every cross-section

shall be as nearly as possible the same as at the cross-section which traverses the rivet-holes. Thus, while the strength of the piece against a dead pull is kept unimpaired, its extensibility, and consequently its resilience, are considerably increased.

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PLAN OF THE BRITISH & FOREIGN STEAM NAVIGATION CO. LTD. STEAMER

PLAN OF THE BRITISH & FOREIGN STEAM NAVIGATION CO. LTD. STEAMER

PLATE I



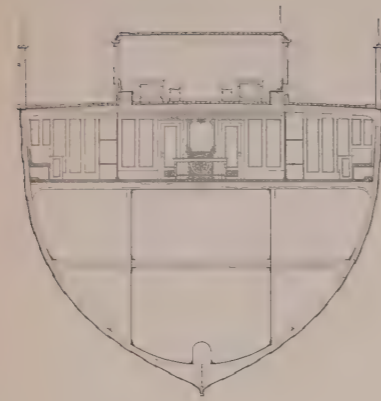
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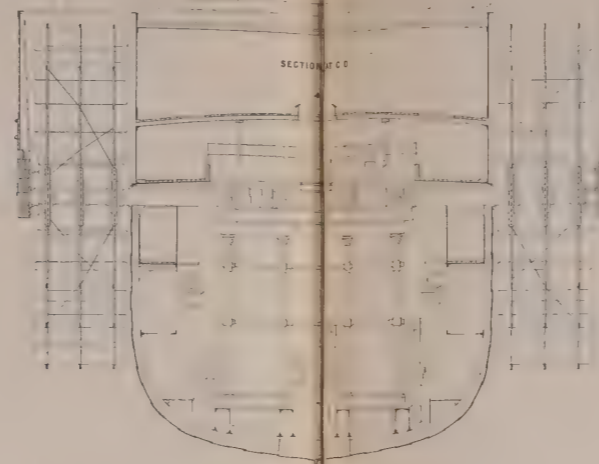
PLATE A

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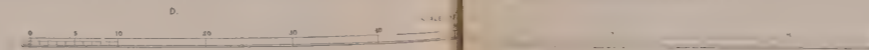
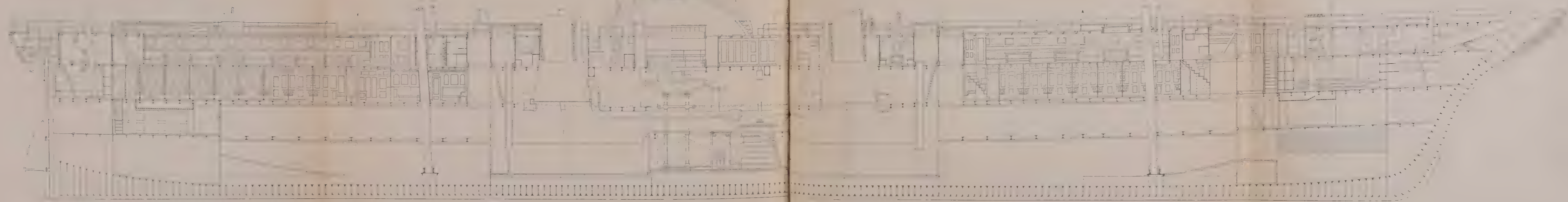
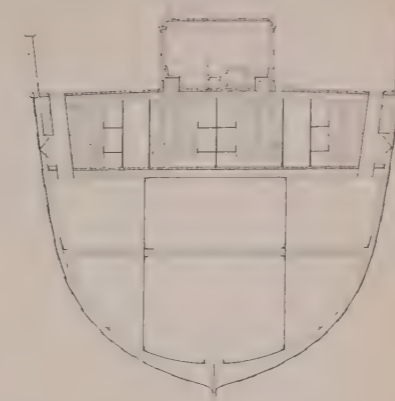
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SECTION AT C.D.

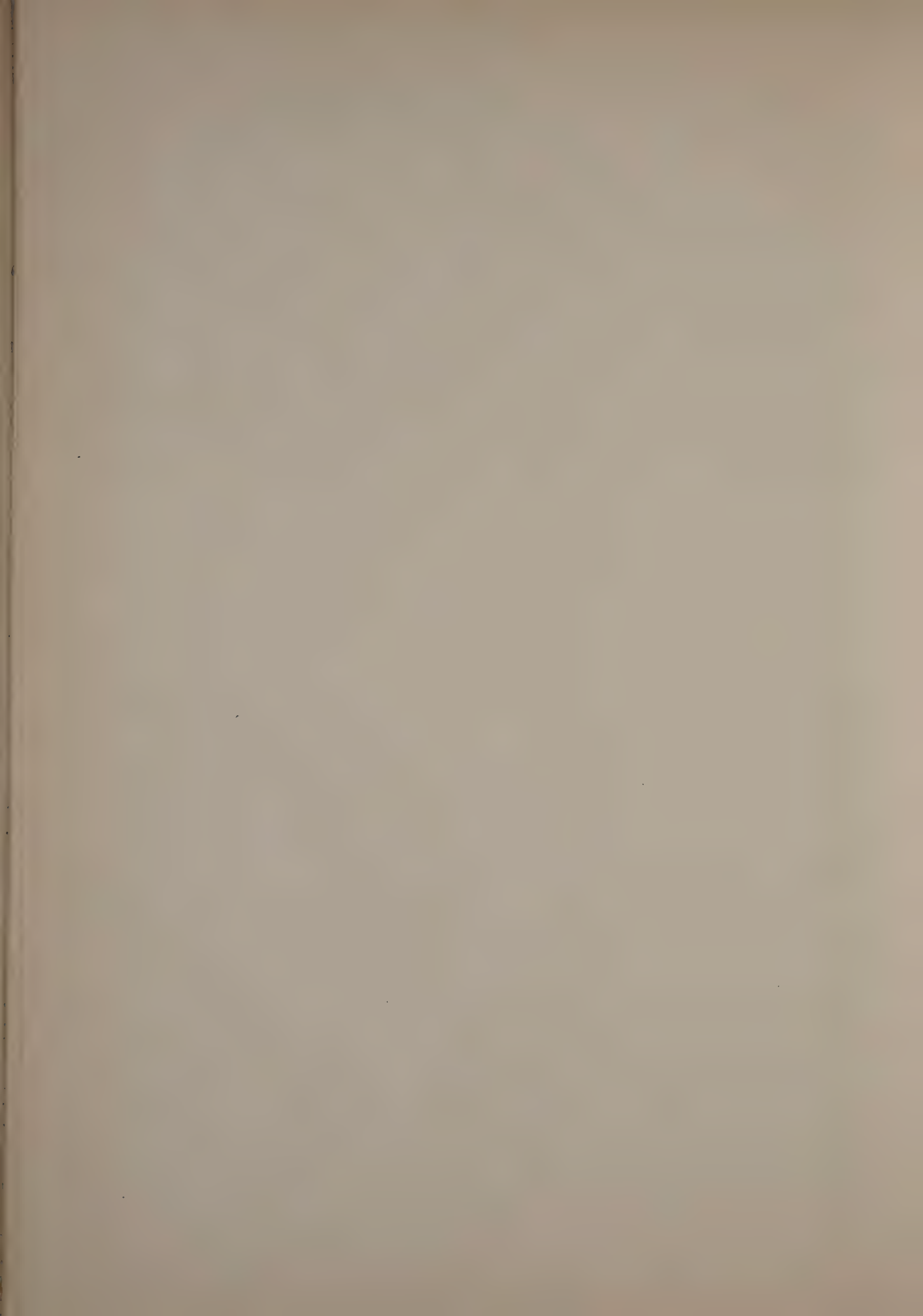


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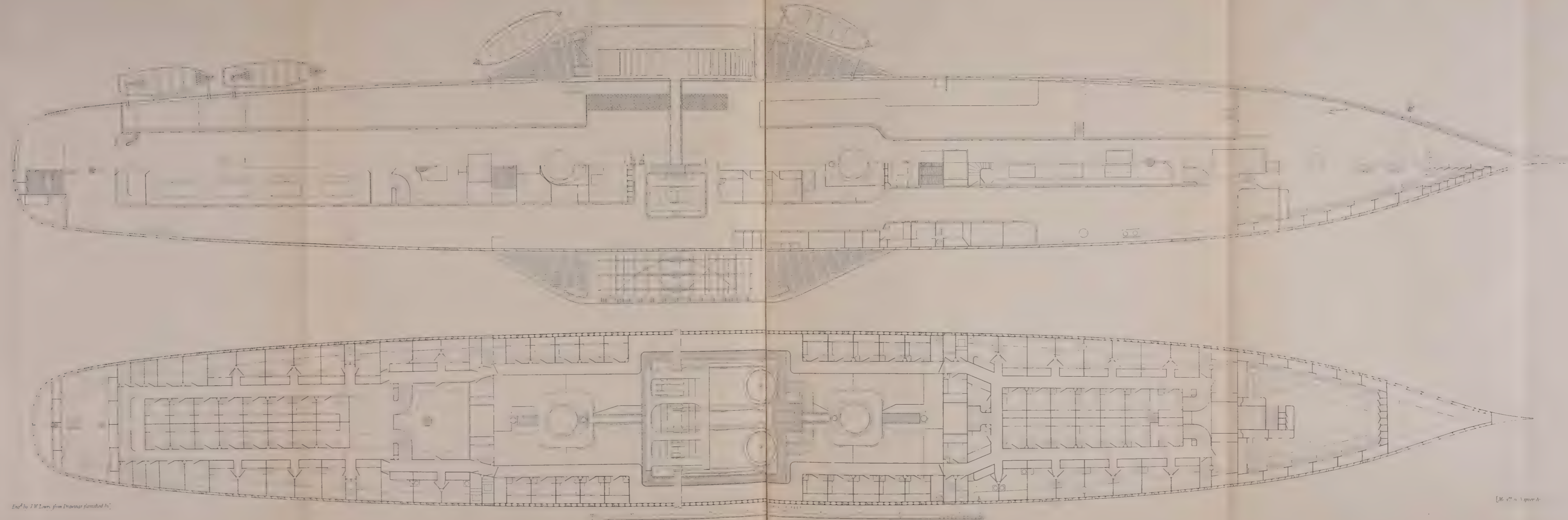
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DECK PLANS OF THE BRITISH & NORTH AMERICAN ROYAL MAIL STEAM SHIP "PERSIA."

PLATE 3.

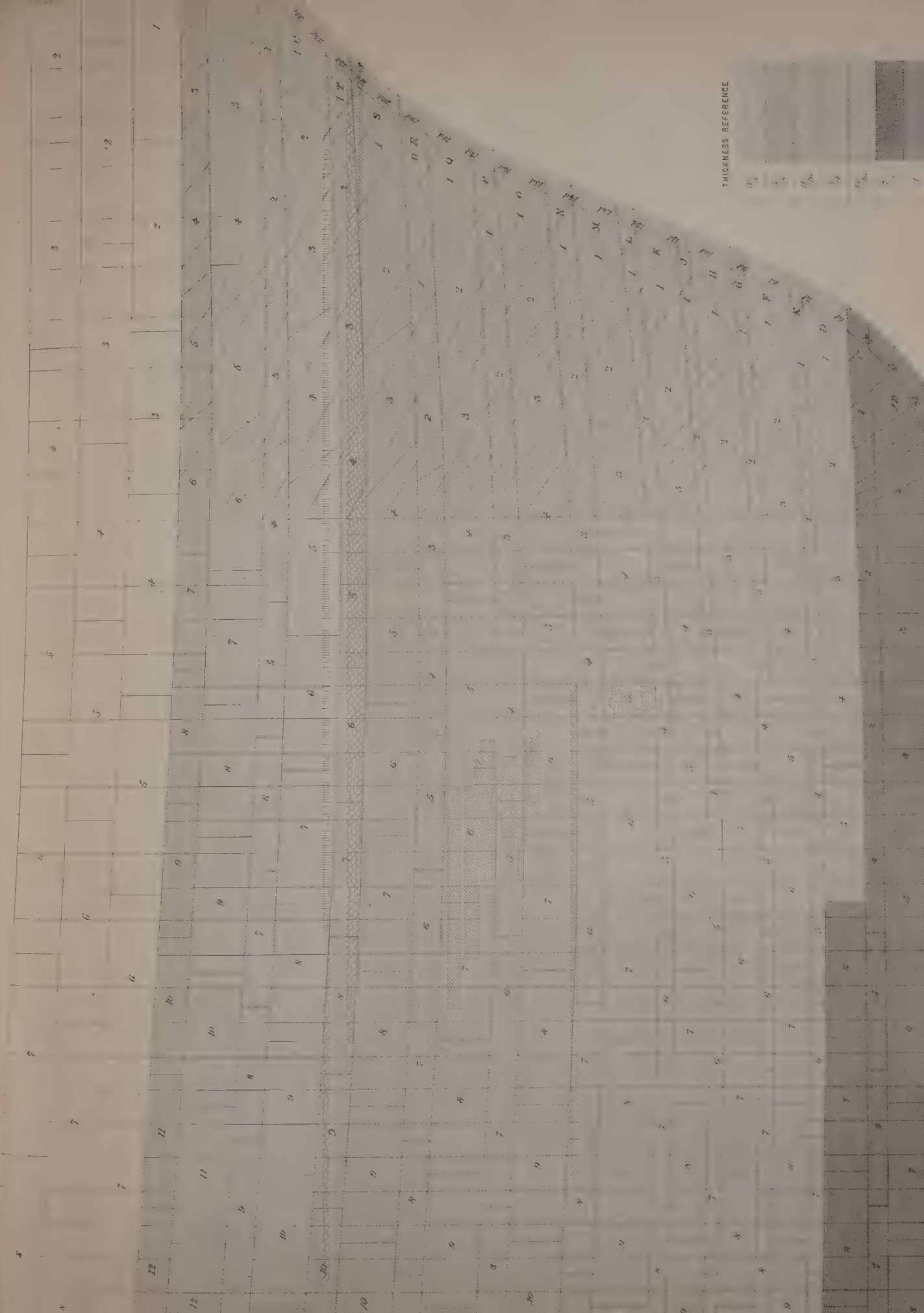
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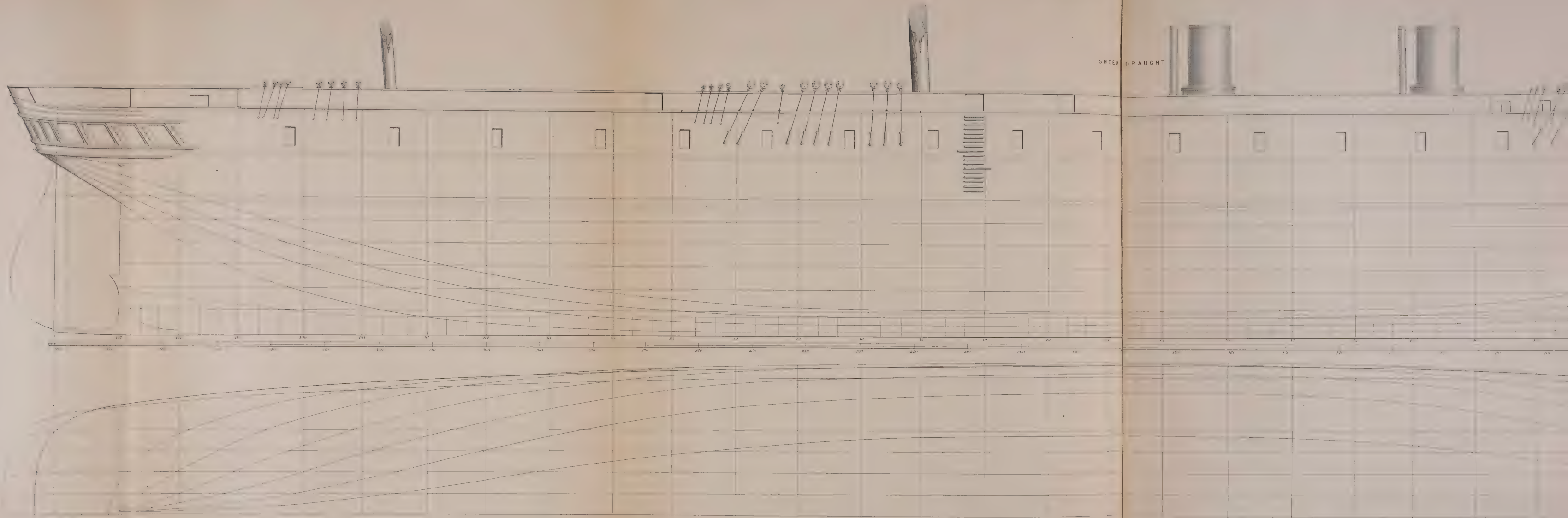
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[M. 22 n. 3 p. 10]



HER MAJESTY'S IRON, IRON CLAD STEAM BATTLESHIP "VICTORIA" OF 1250 HORSES POWER.

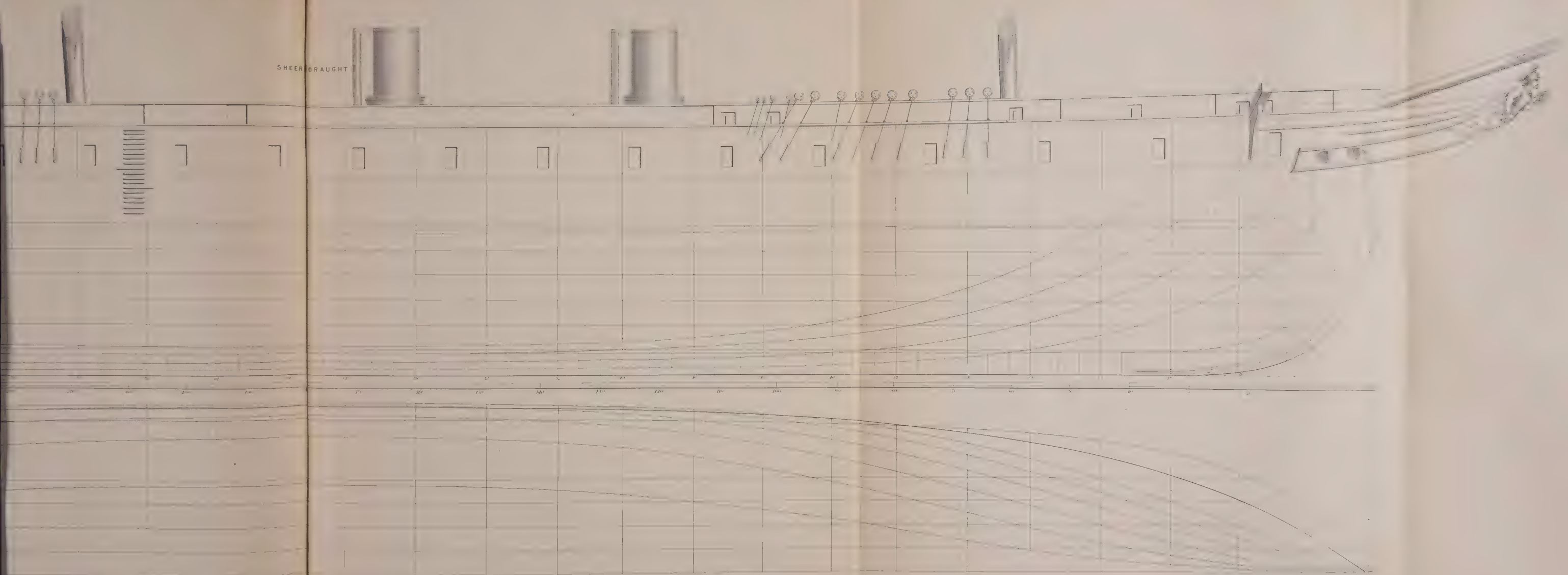
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SHEER DRAUGHT

HALF BREADTH PLAN

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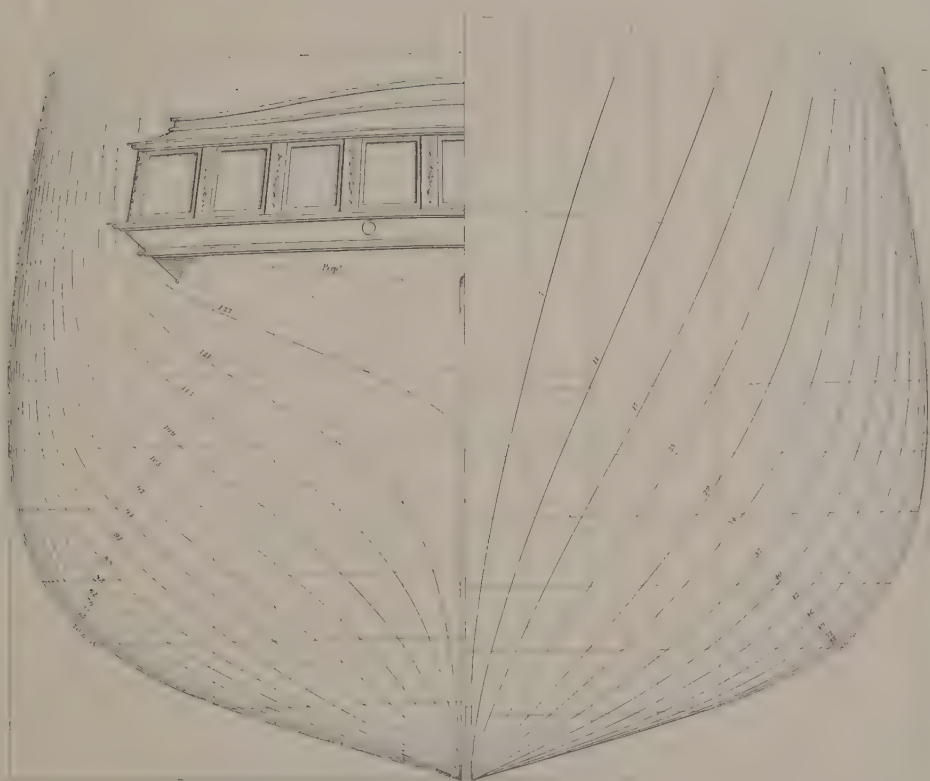
SHEER DRAUGHT

HALF BREADTH PLAN

PRINCIPAL DIMENSIONS.

	<i>Fe.</i>	<i>In.</i>
LENGTH BETWEEN THE PERPENDICULARS.....	380.	0.
" OF THE KEEL FOR TONNAGE.....	337.	57.
BREADTH EXTREME.....	58.	0.
" FOR TONNAGE.....	58.	0.
DEPTH IN HOLD.....	21.	1.
BURTHEN IN TONS. O. M. N ^o 6038	$\frac{85}{100}$	

BODY PLAN.

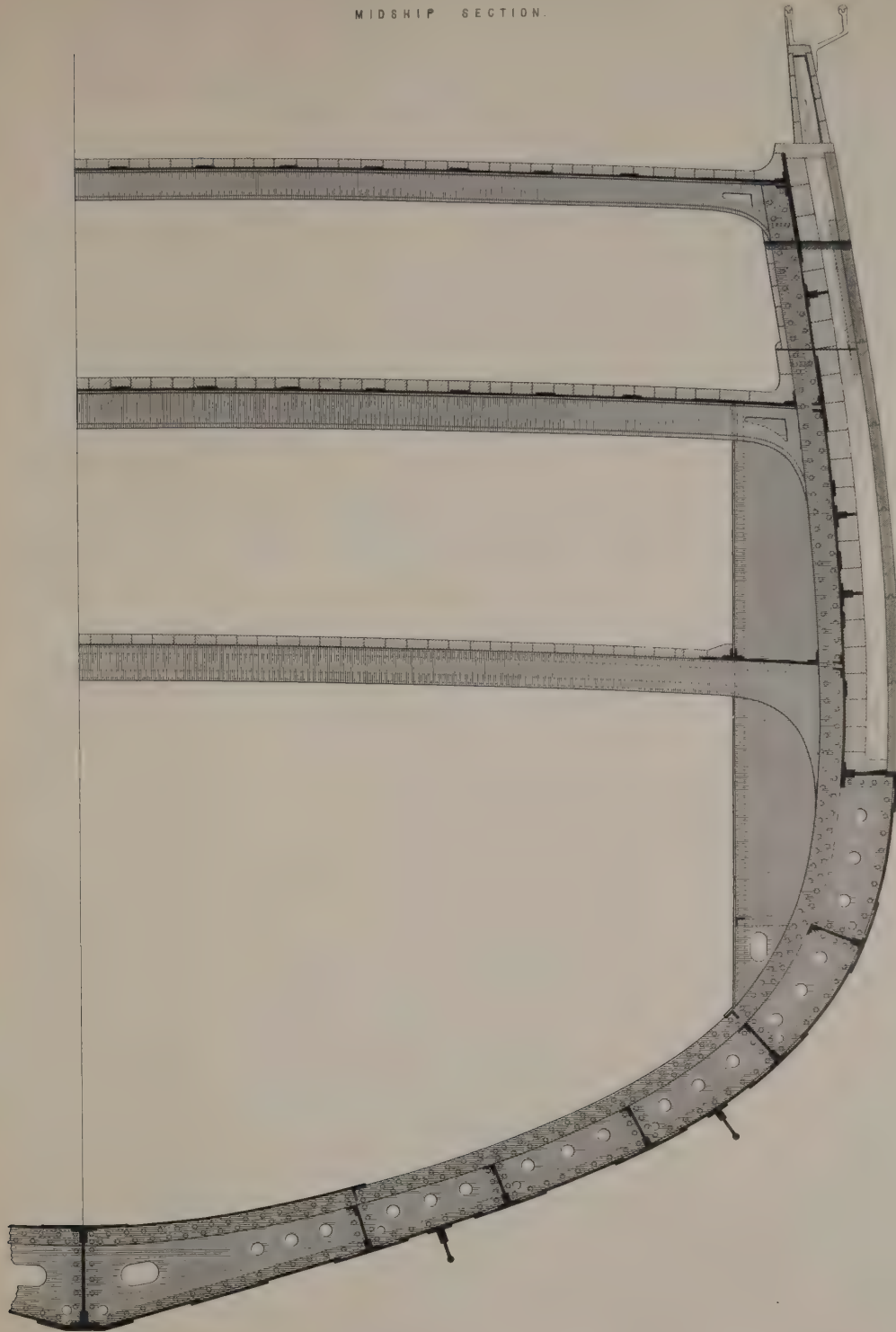


	LAUNCHED.		LOADED.	
	<i>Fe.</i>	<i>In.</i>	<i>Fe.</i>	<i>In.</i>
DRAUGHT OF WATER FORWARD	15	3.	25	10.
" " " AFT	17	3.	26	9.
DISPLACEMENT IN TONS	1350.		8997.	
SPEED IN KNOTS, ON TRIAL	14.350			
INDICATED HORSE-POWER.	5471			

HER MAJESTY'S IRON, IRON CLAD SHIP "WARRIOR."

PLATE B.
5

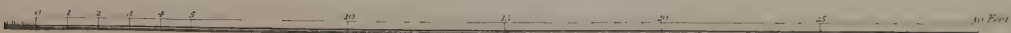
MIDSHIP SECTION.



PLAN AT HEIGHT OF PORT SILL.



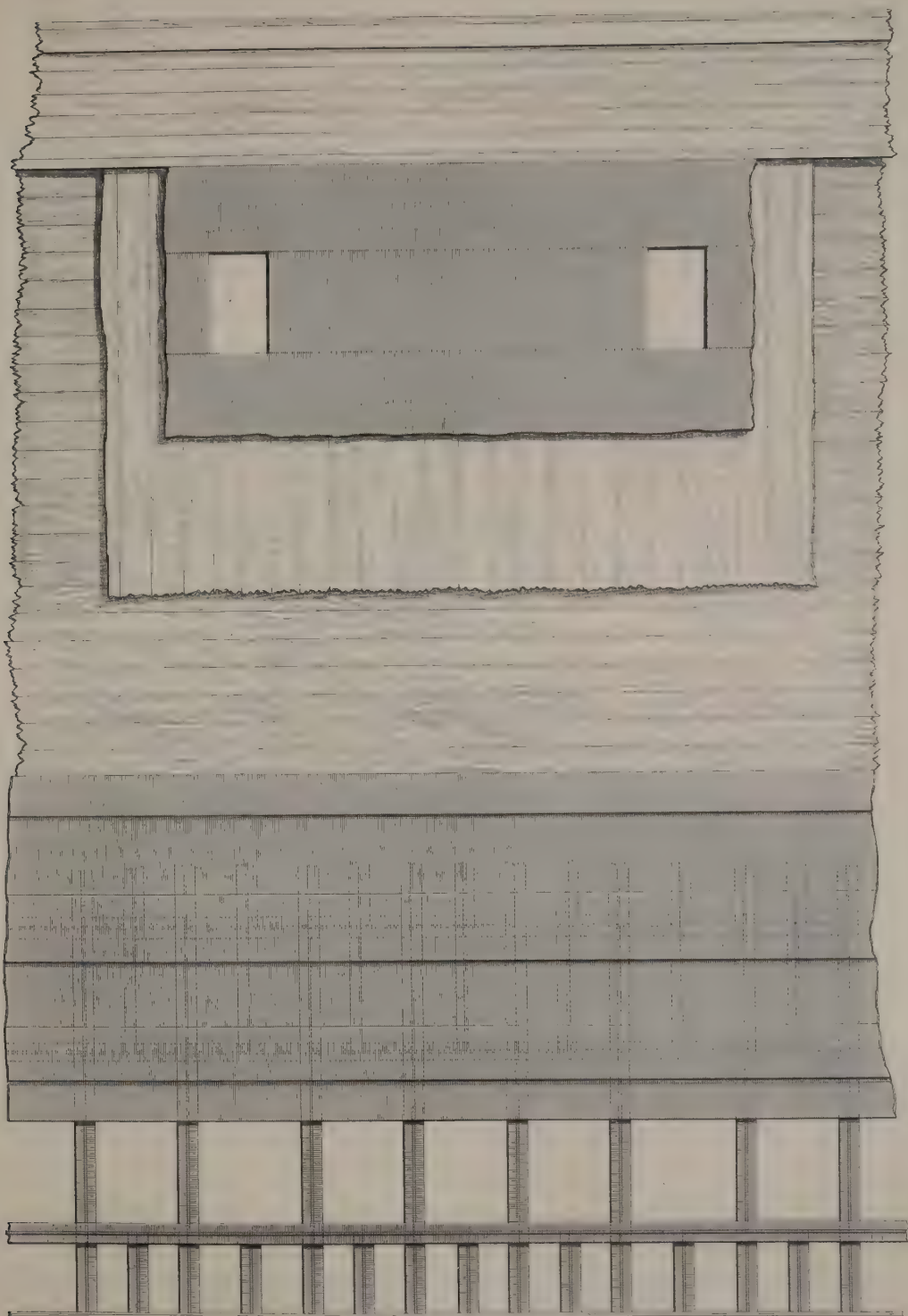
SCALE OF FEET



HER MAJESTY'S IRON, IRON CLAD SHIP "WARRIOR."

PLATE B.
4

EXTERNAL VIEW

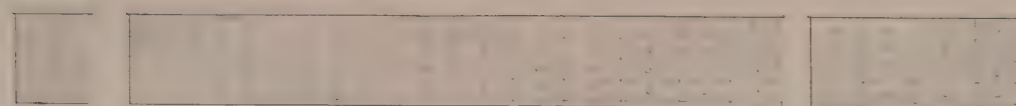


SCALE OF FEET

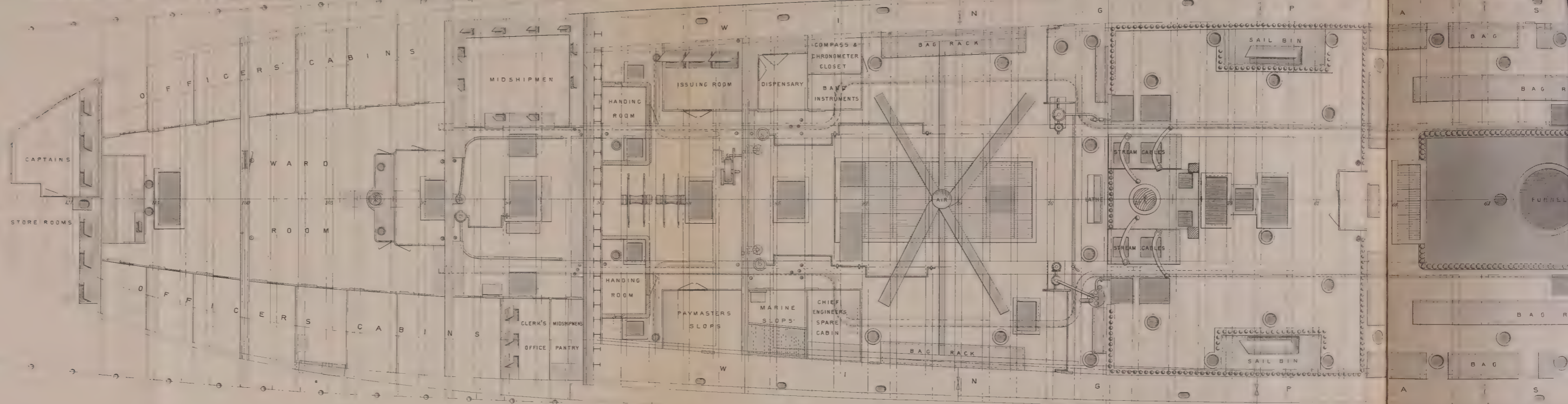


HER MAJESTY'S IRON, IRON CLAD SHIP "WARRIOR" OF 1860

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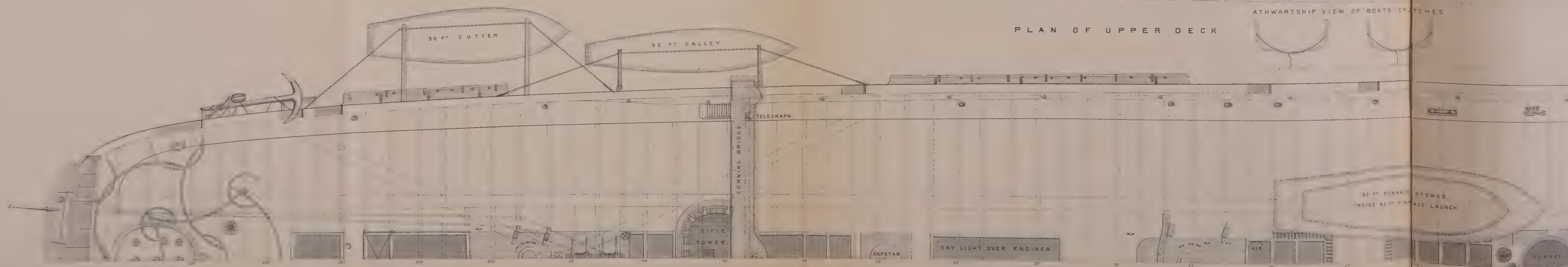


POSITION OF WASH STANDS AND DRAWERS IN SUN ROOM BY 1/4 SCALE

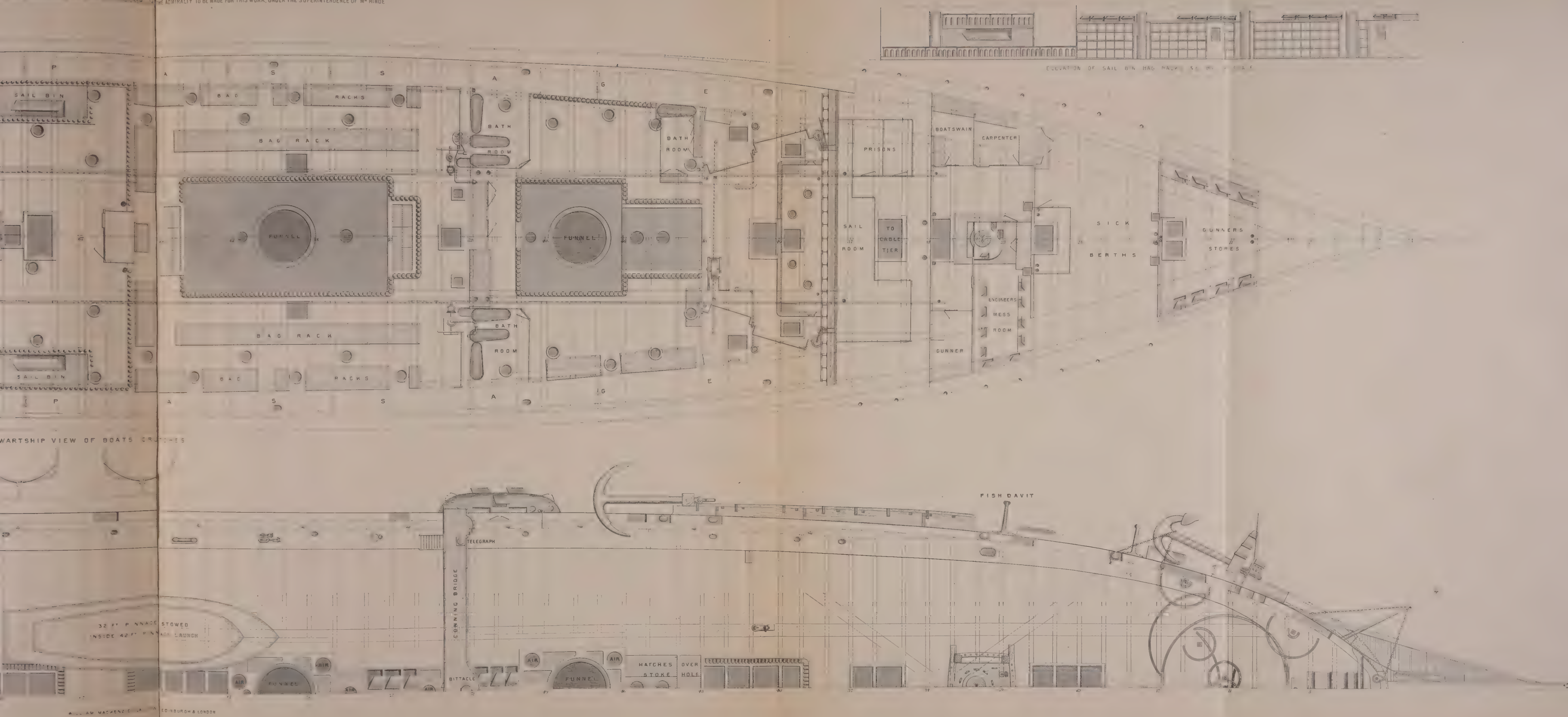


PLAN OF UPPER DECK

ATHWARTSHIP VIEW OF BOATS CRUTCHES



DESIGNED BY THE LORDS COMMANDERS OF THE ADMIRALTY TO BE MADE FOR THIS WORK, UNDER THE SUPERINTENDENCE OF M^r HINDS



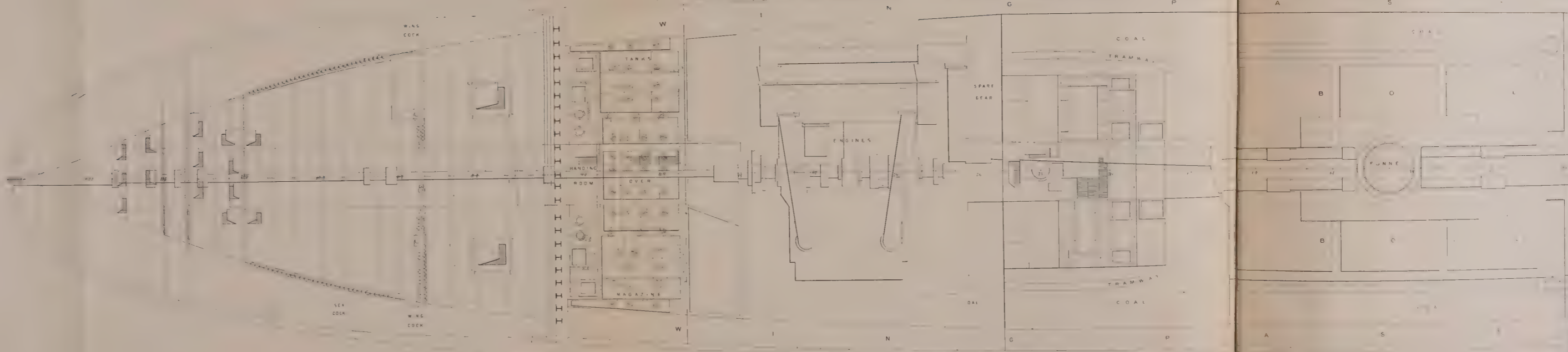
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SECTION AT HANDING ROOM

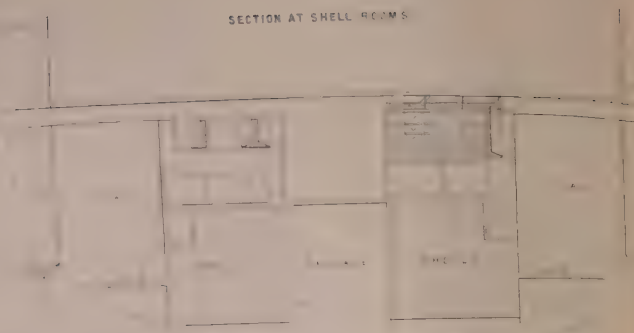
SECTION AT CENTRE OF MAGAZINE

SECTION AT SHELL ROOMS

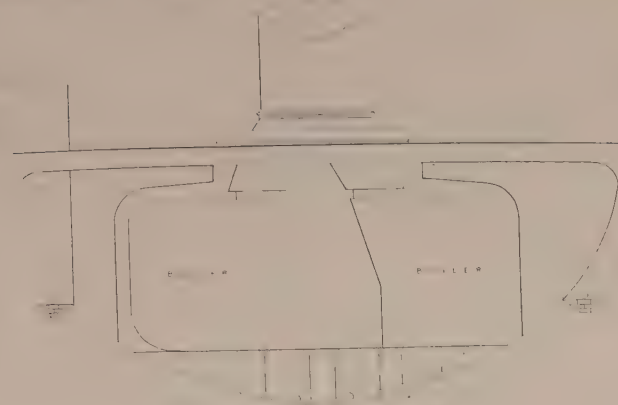
PLAN OF HOLD



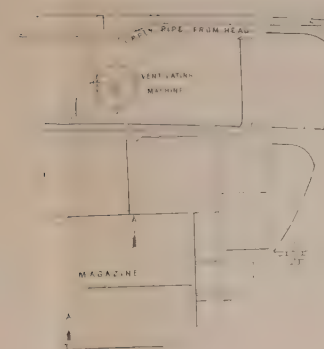
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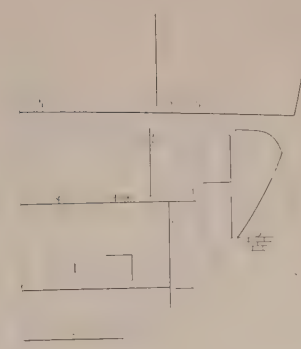
SECTION AT FORE END OF BOILERS



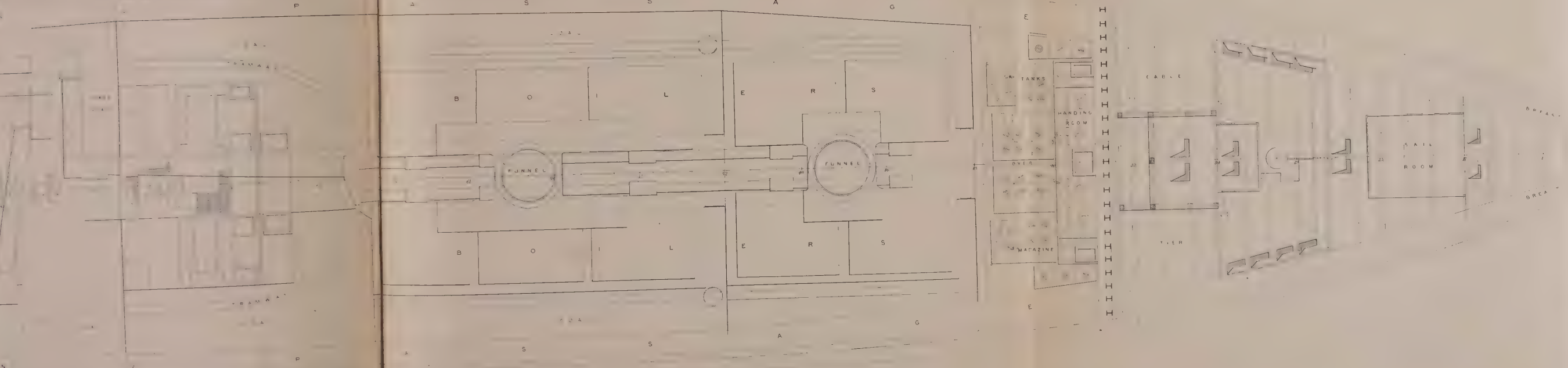
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SECTION AT HANDING ROOM



PLAN OF HOLD

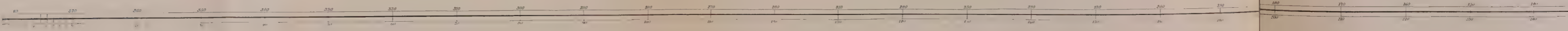
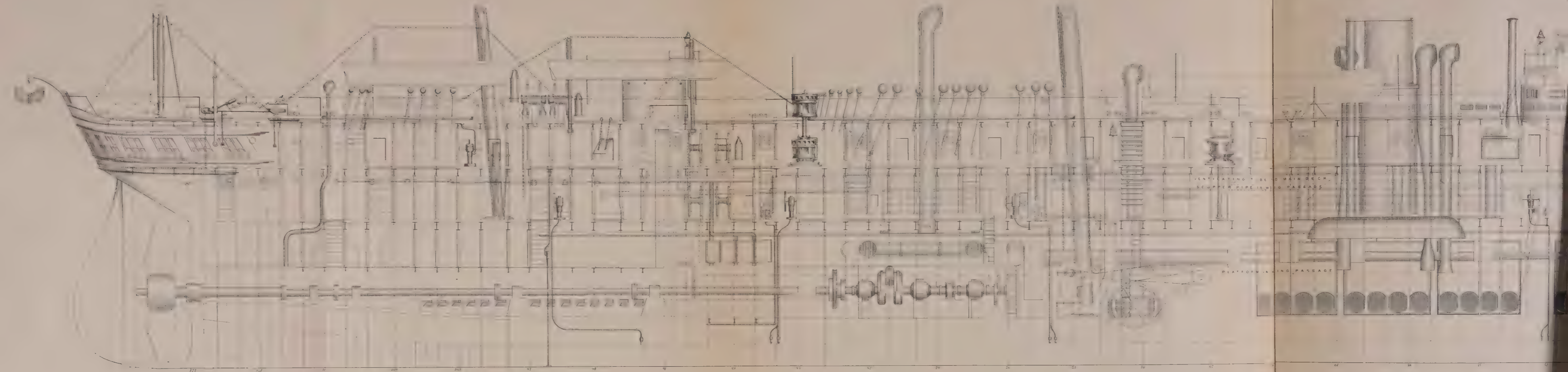




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PROFILE OF THE IRON CLAD WORKS &c.

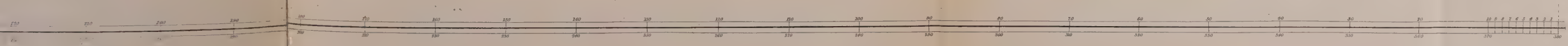
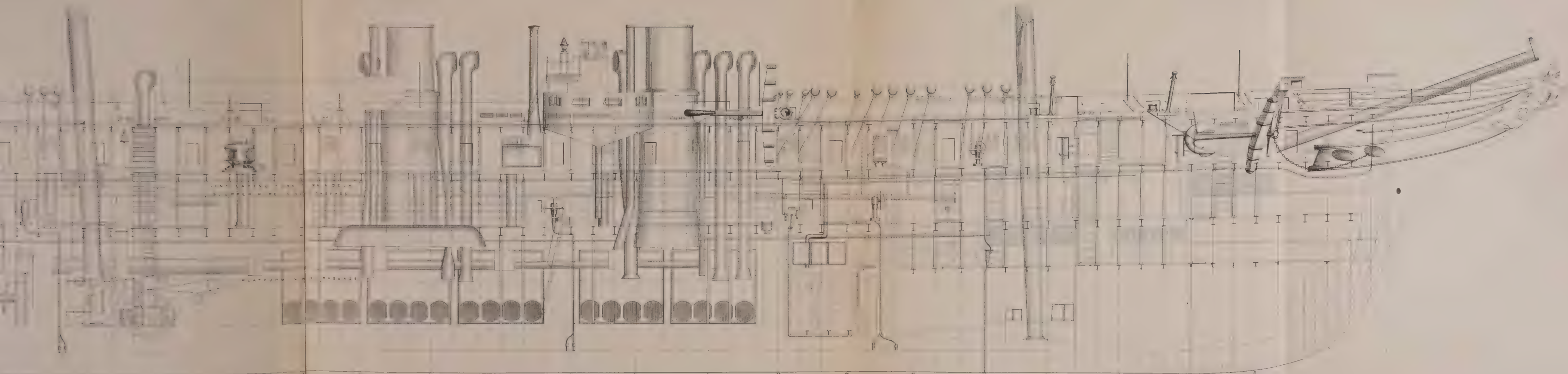


MAJESTY'S IRON, IRON CLAD SHIP 'WARRIOR' OF 1250 HORSES POWER.

PLATE B

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PROFILE OF THE SHIP

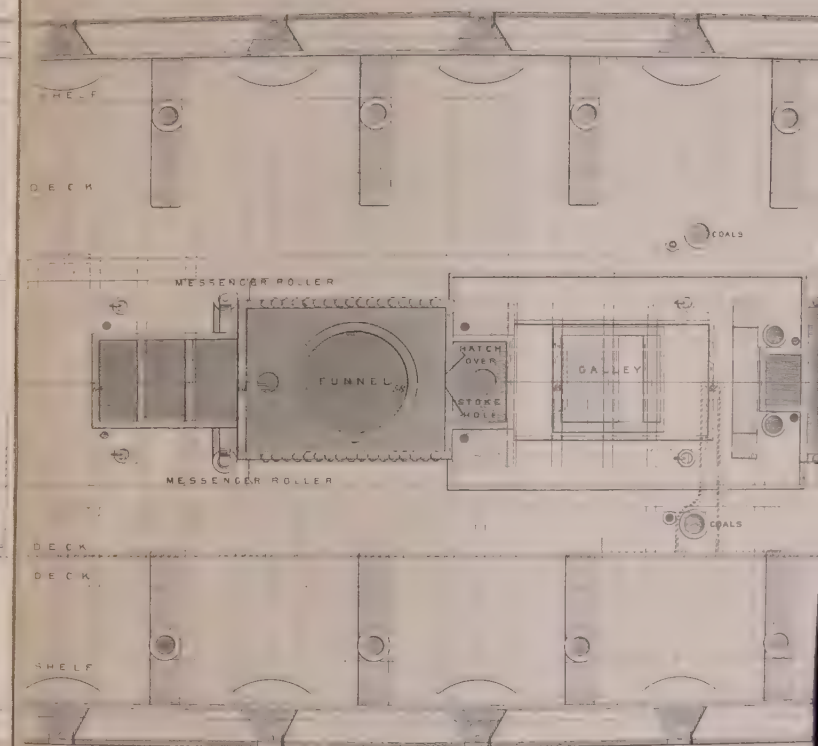
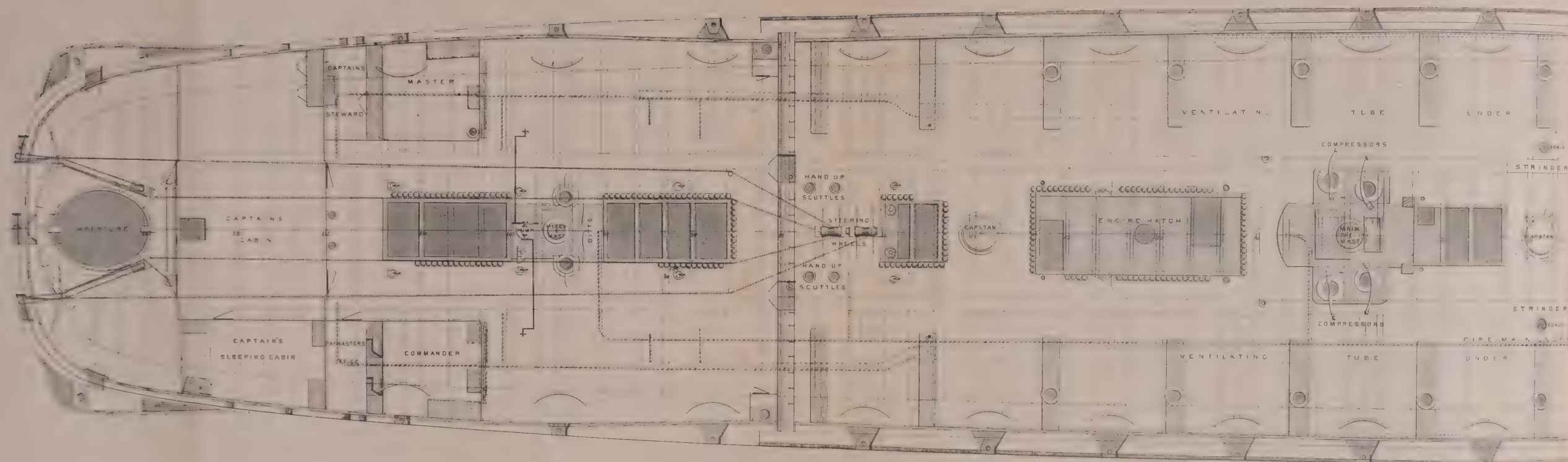


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PLAN OF MAIN DECK



0 10 20 30 40 50 60 FEET

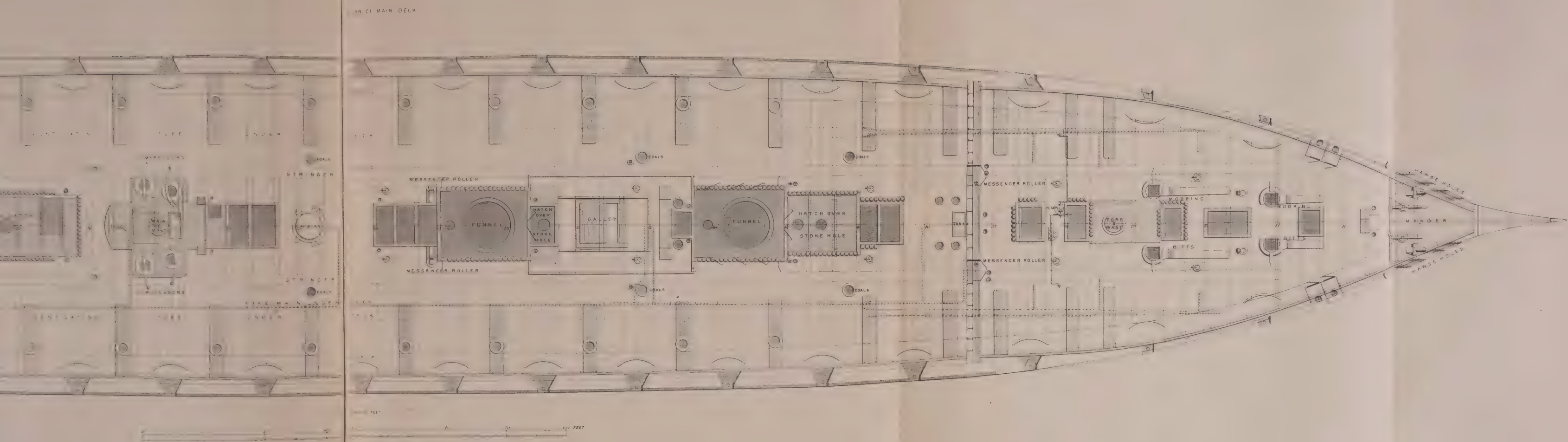
0 10 20 30 40 50 60 FEET

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PLATE B

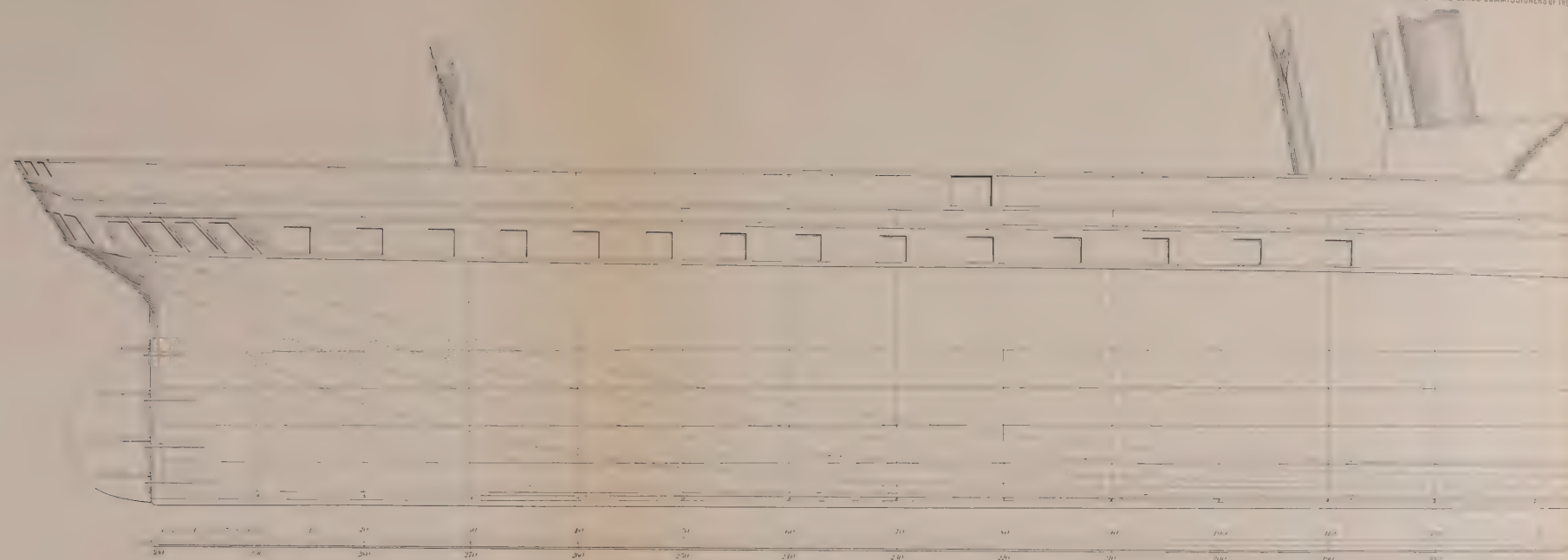


WILLIAM MACKENZIE DUNDEE EDINBURGH & LONDON

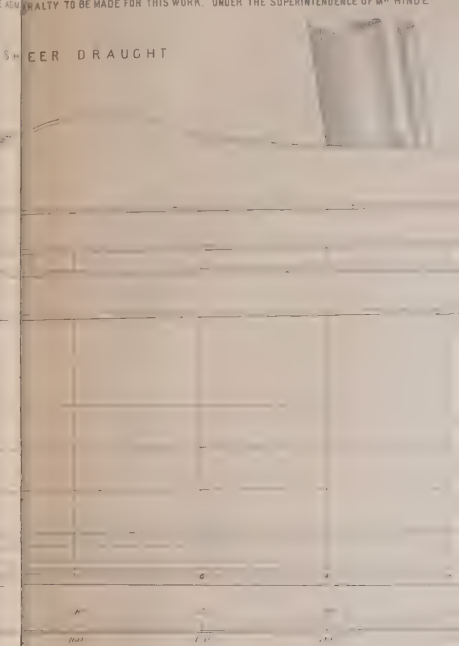
HER MAJESTY'S YACHT 'VICTORIA' & 'ALBERT' OF 600 HORSES POWER

ENGRAVED FROM OFFICIAL DRAWINGS AUTHORIZED BY THE LORDS COMMISSIONERS OF THE ADMIRALTY TO BE MADE FOR THIS WORK, UNDER THE SUPERINTENDENCE OF MR. HINDE

BODY PLAN



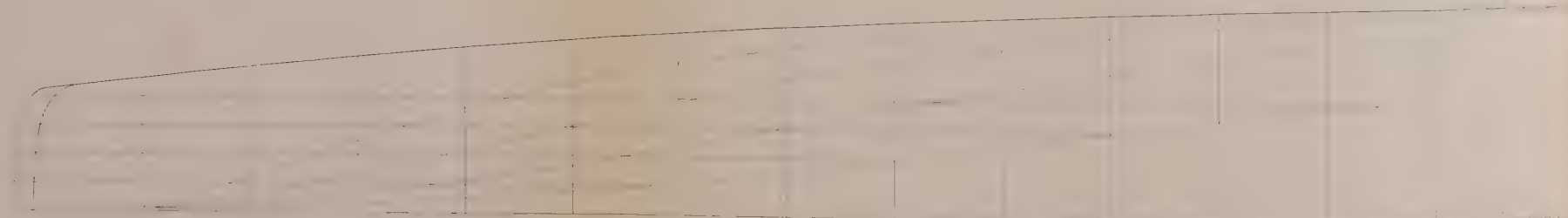
SWEEP DRAUGHT



PRINCIPAL DIMENSIONS

LENGTH BETWEEN THE PERPENDICULARS	117' 0"
OF THE KEEL FOR TONNAGE	117' 0"
BREADTH EXTREME	26' 0"
FOR TONNAGE	26' 0"
MOULDED	26' 0"
DEPTH IN HOLD	12' 0"
BURTHEM IN TONS	1,100

HALF BREADTH PLAN



WILLIAM MACFARLANE & CO. LONDON

WILLIAM MACFARLANE & CO. LONDON

THE BRITISH STEAM NAVY
DESIGNED BY THE ADMIRALTY
UNDER THE SUPERINTENDENCE OF MR. H. K. WATKINS

PLATE C
1

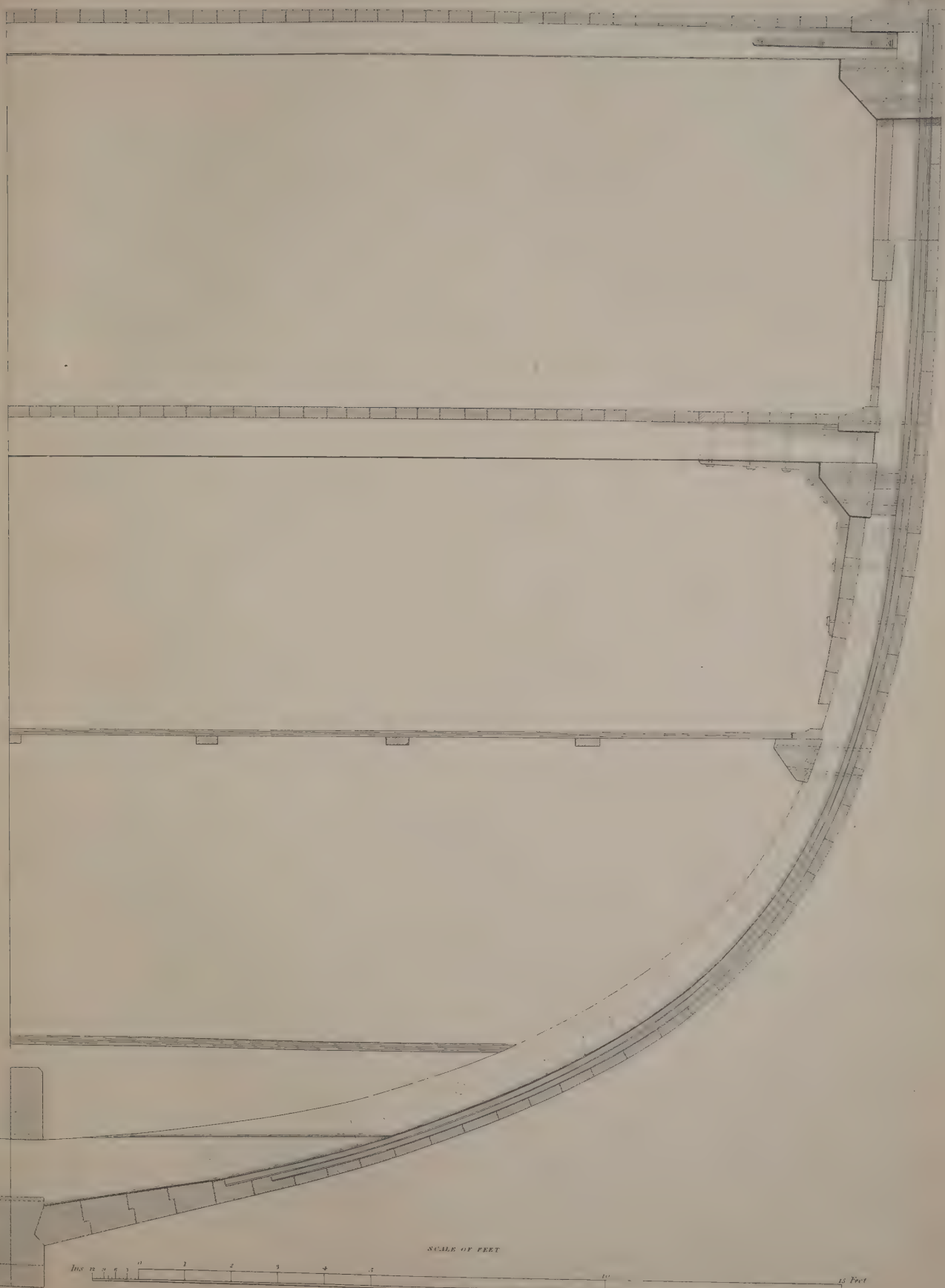
PLAN OF THE
SHIP'S DRAUGHT

HALF-BREADTH PLAN

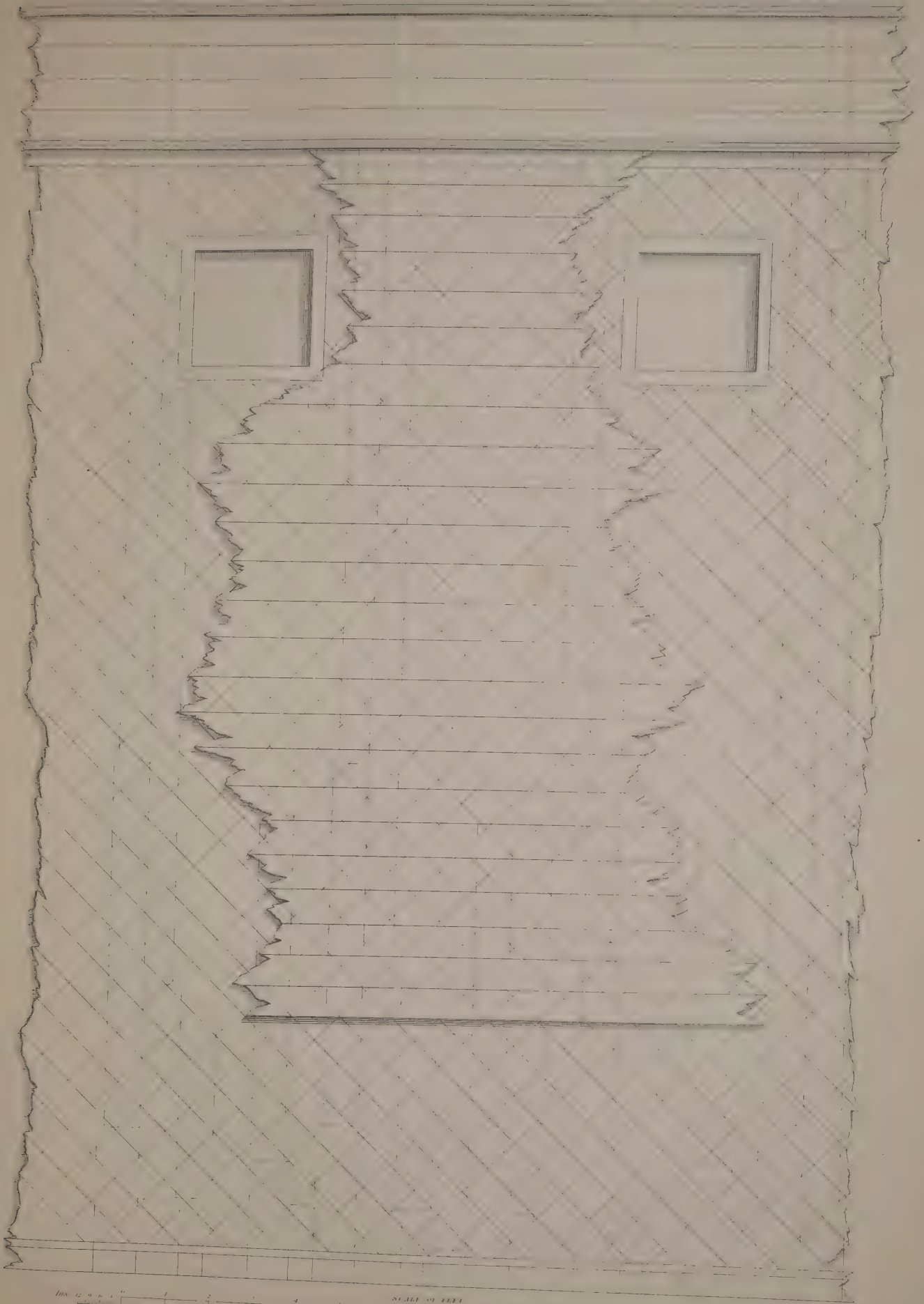
WILLIAM BATHENZ
DRAWN BY

EDINBURGH & LONDON

MIDSHIP SECTION.



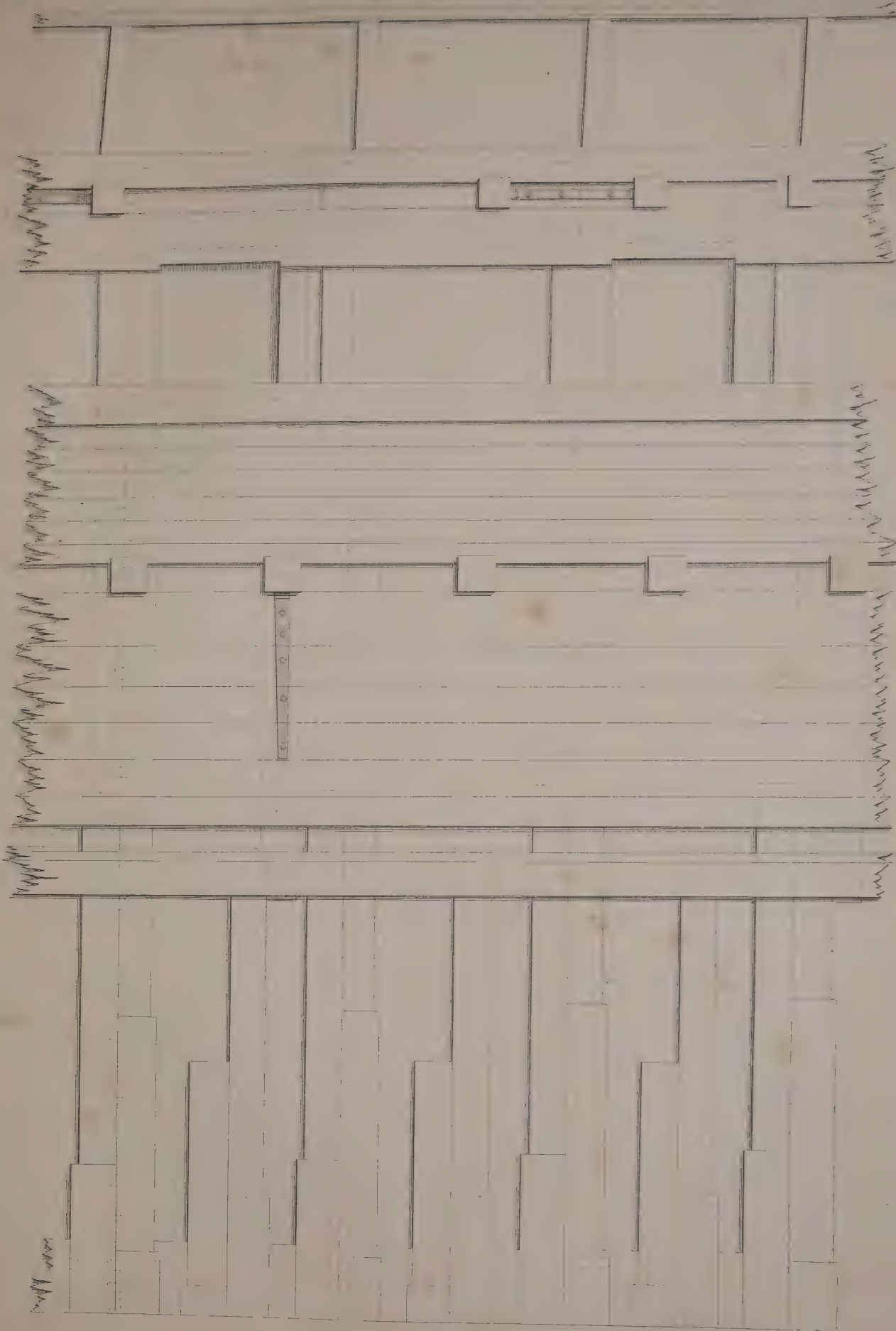
EXTERNAL VIEW OF PLANKING.



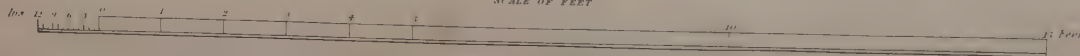
100 12 0 6 1 10 1 2 1 4

SCALE OF FEET

EX. 100. 12. 0. 6. 1. 10. 1. 2. 1. 4.



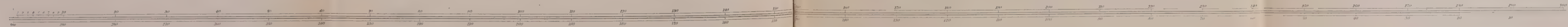
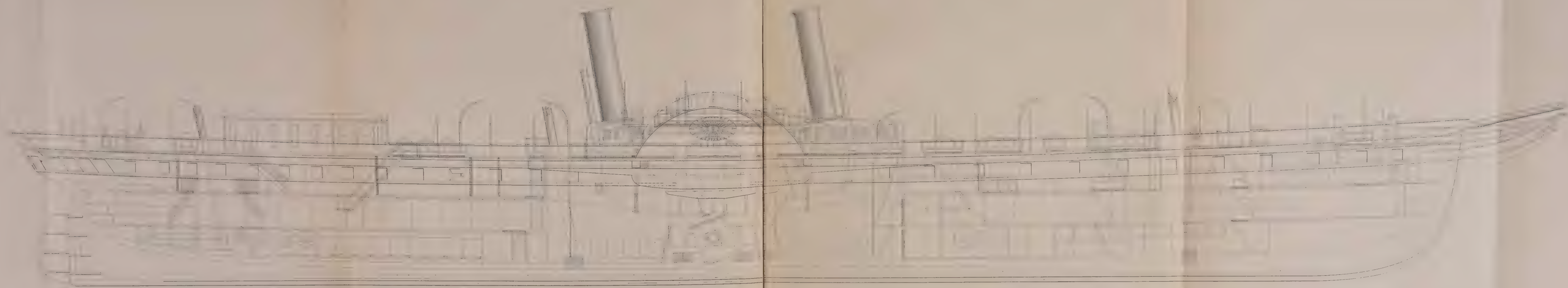
SCALE OF FEET





PROFILE OF INBOARD WORKS OF HER MAJESTY'S YACHT "VICTORIA & ALBERT" OF 600 HORSES POWER.

ENGRAVED FROM OFFICIAL DRAWINGS AUTHORIZED BY THE LORDS COMMISSIONERS OF THE ADMIRALTY TO BE MADE FOR THIS WORK UNDER THE SUPERINTENDENCE OF MR HINDS.



WILLIAM MACKENZIE, GLASGOW.

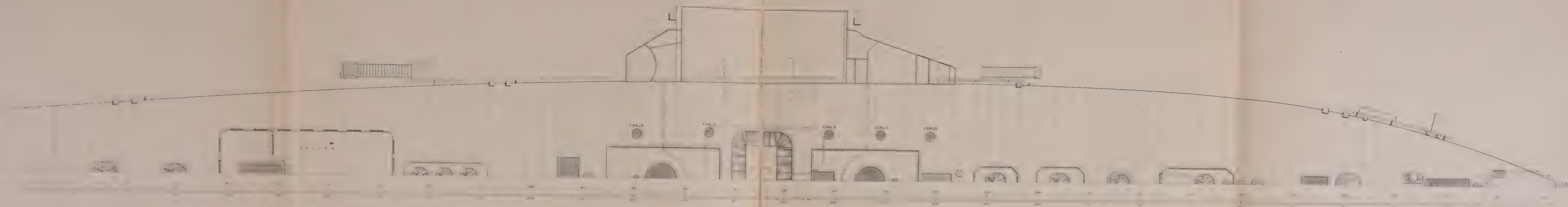


H. M. J. YACHT "VICTORIA" A. A. ALBERT OF 300 HORSE POWER.

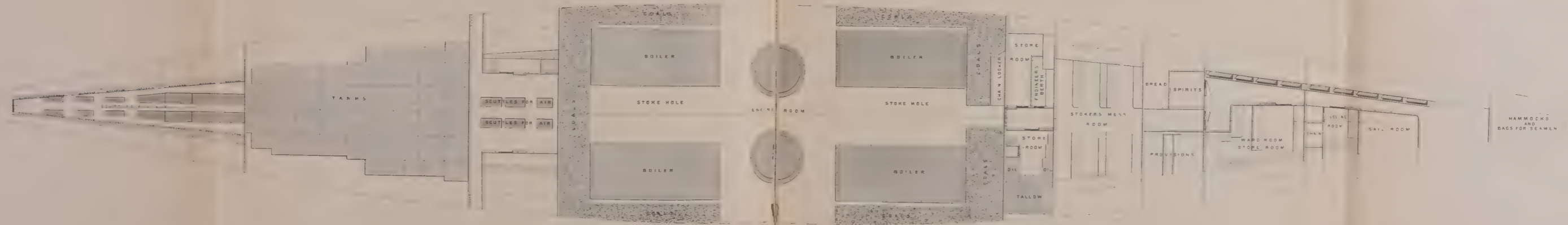
ENGRAVED FROM OFFICIAL DRAWINGS AUTHORIZED BY THE LORDS COMMISSIONERS OF THE ADMIRALTY TO BE MADE FOR THIS WORK UNDER THE SUPERINTENDENCE OF M^r HINDS.

PLATE I

PLAN OF UPPER DECK



PLAN OF HOLD



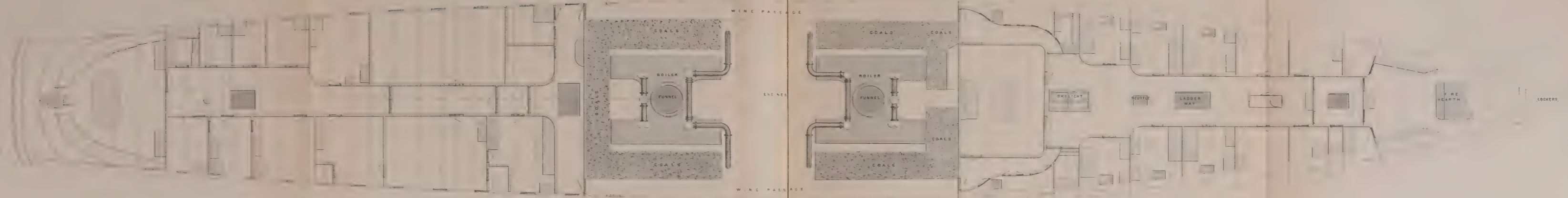
WILLIAM MACKENZIE, GLASGOW, EDINBURGH & LONDON

HER MAJESTY'S YACHT VICTORIA & ALBERT OF 800 HORSES POWER.

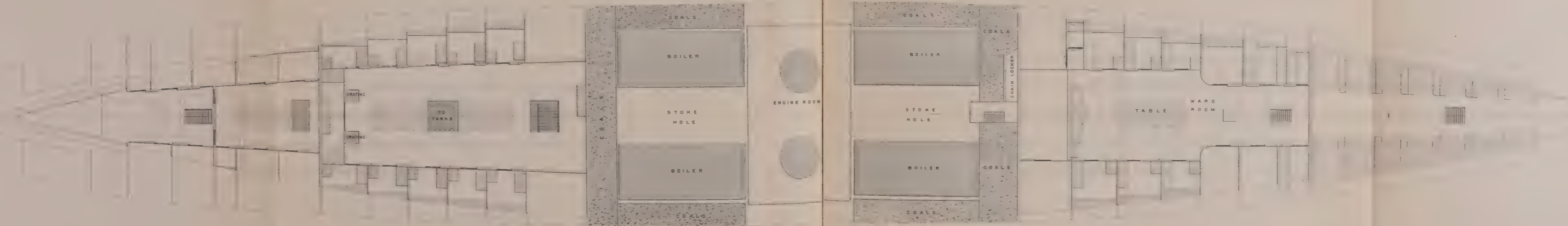
ENGRAVED FROM OFFICIAL DRAWINGS AUTHORIZED BY THE LORDS COMMISSIONERS OF THE ADMIRALTY TO BE MADE FOR THIS WORK. UNDER THE SUPERINTENDENCE OF MR HINDE

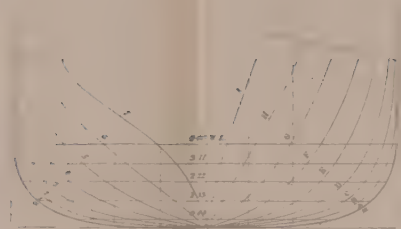
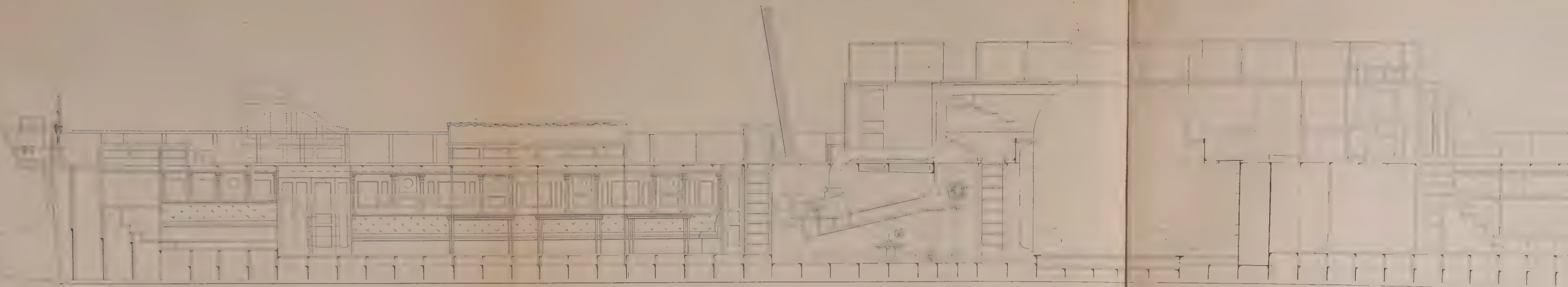
PLATE C

PLAN OF STATE CAIN DECK



PLAN OF LOWER DECK



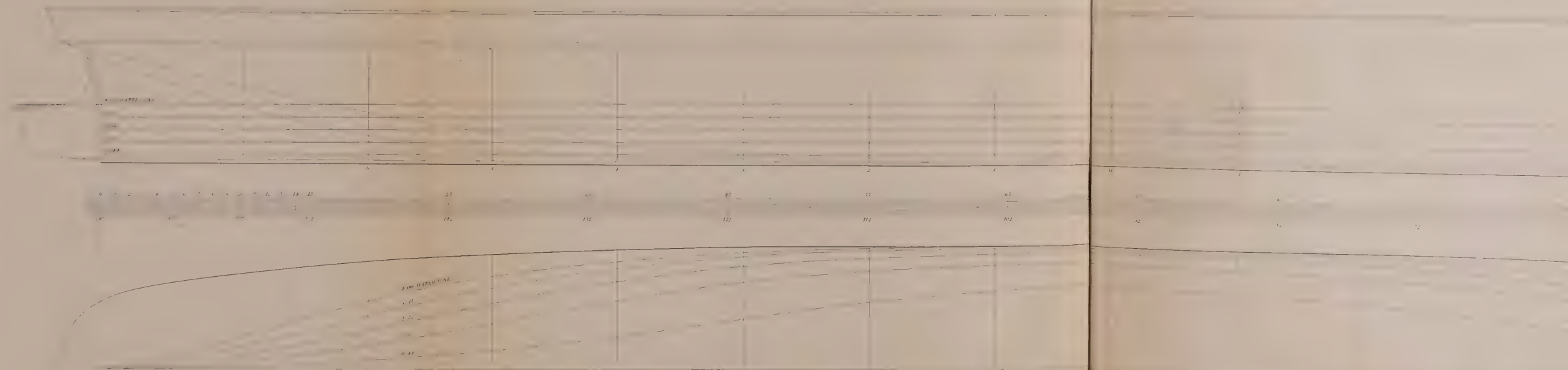


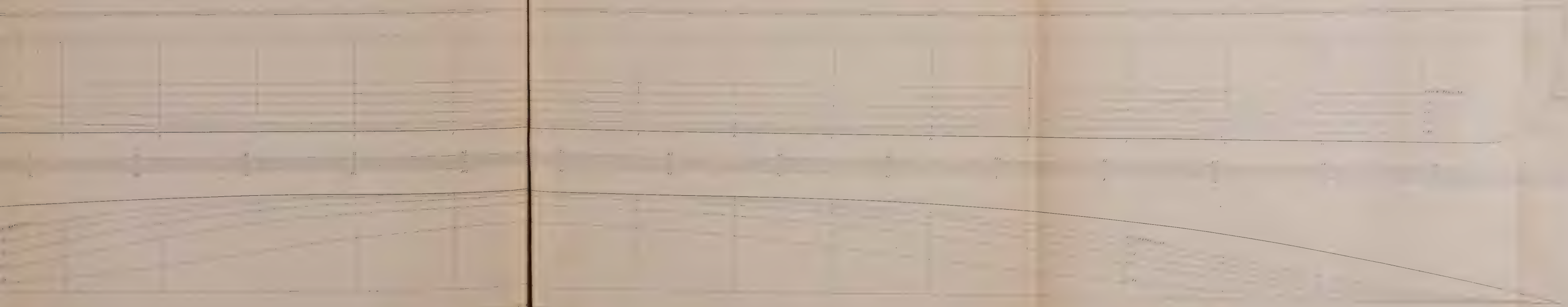
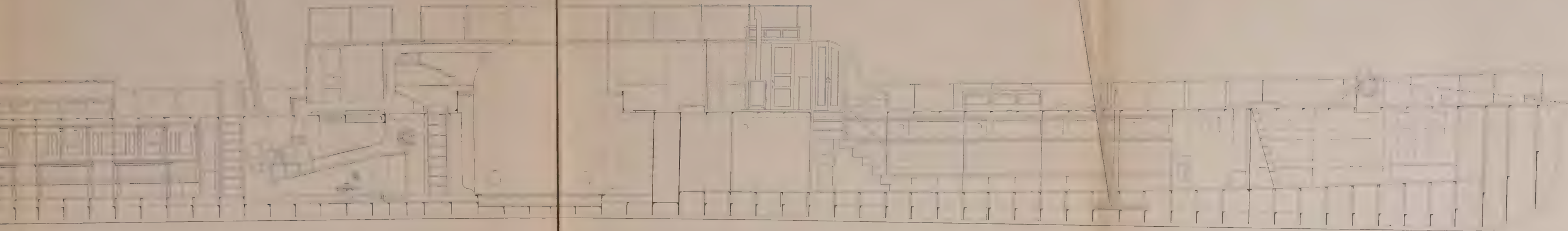
PRINTED BY MESSRS. L. & CO.

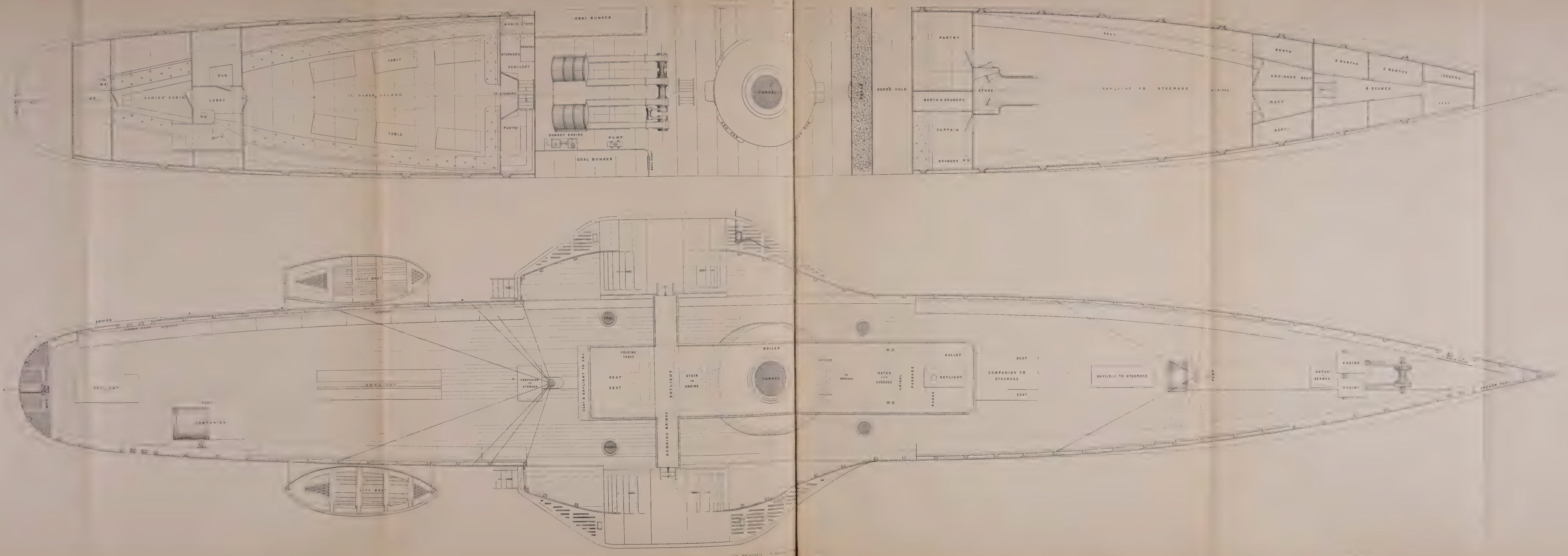
WITH BETWEEN THE ...

GREEN ...

WITH ...







THE IRON STEAM VESSEL 'IONA'

BUILT BY MESSRS J & G THOMSON GLASGOW 1864

PLATE F

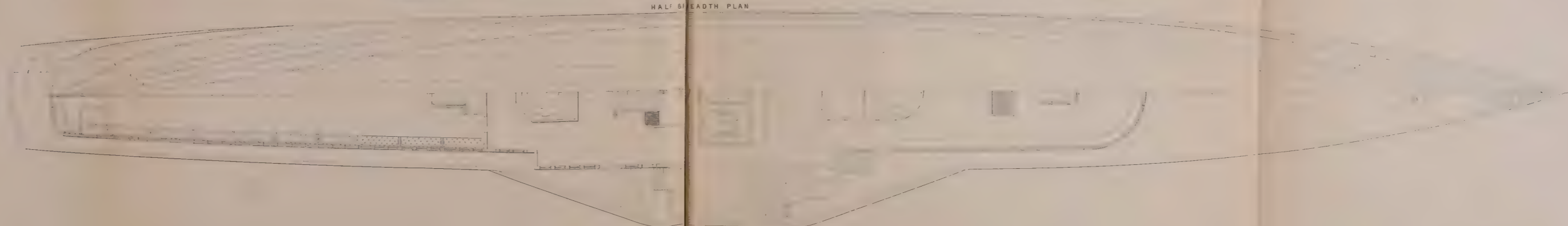
BODY PLAN



ELEVATION & SHEER PLAN

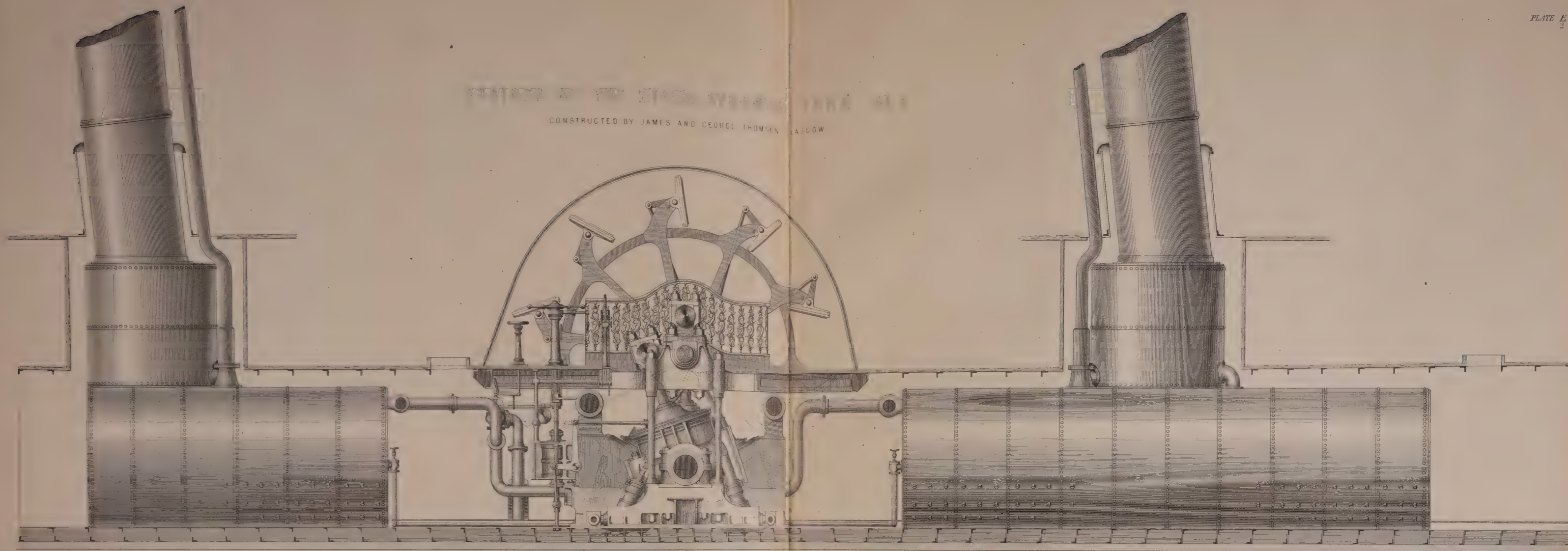


HALF BREADTH PLAN



DECK PLAN

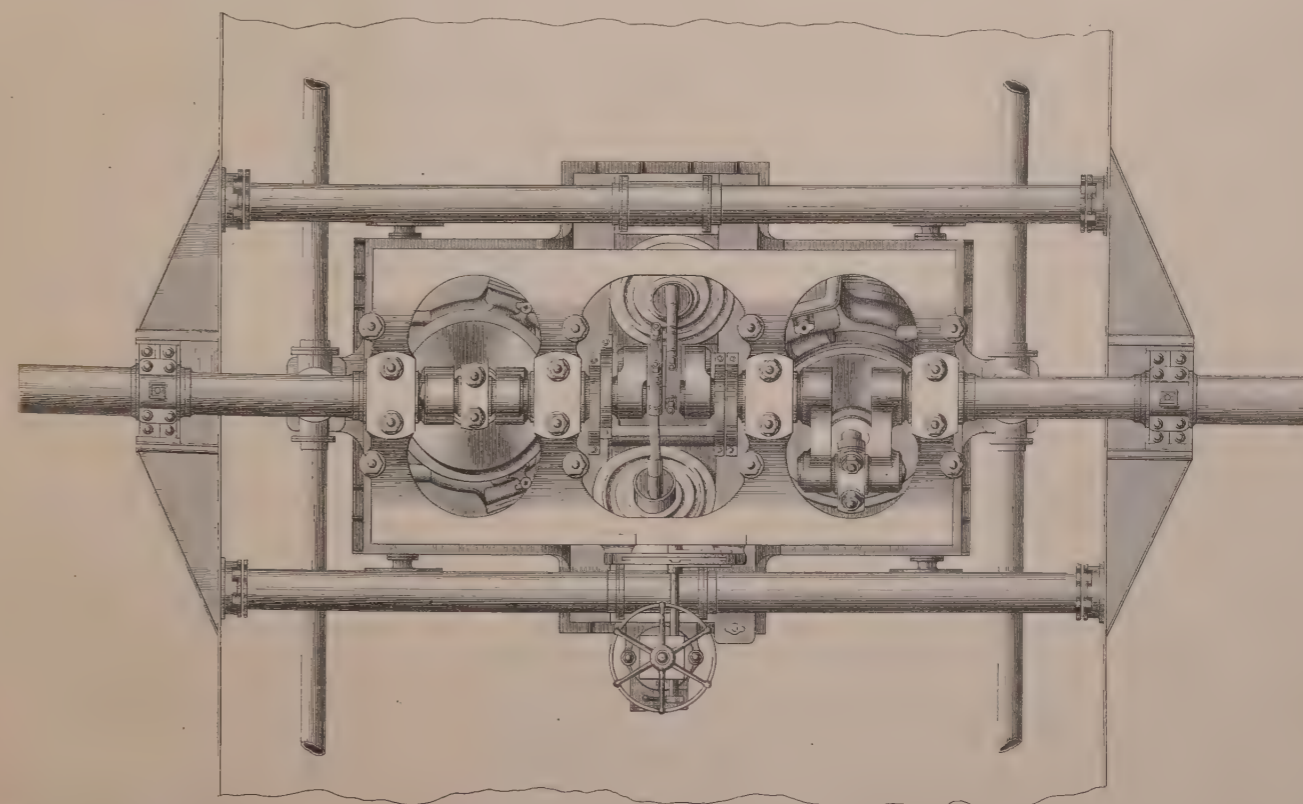
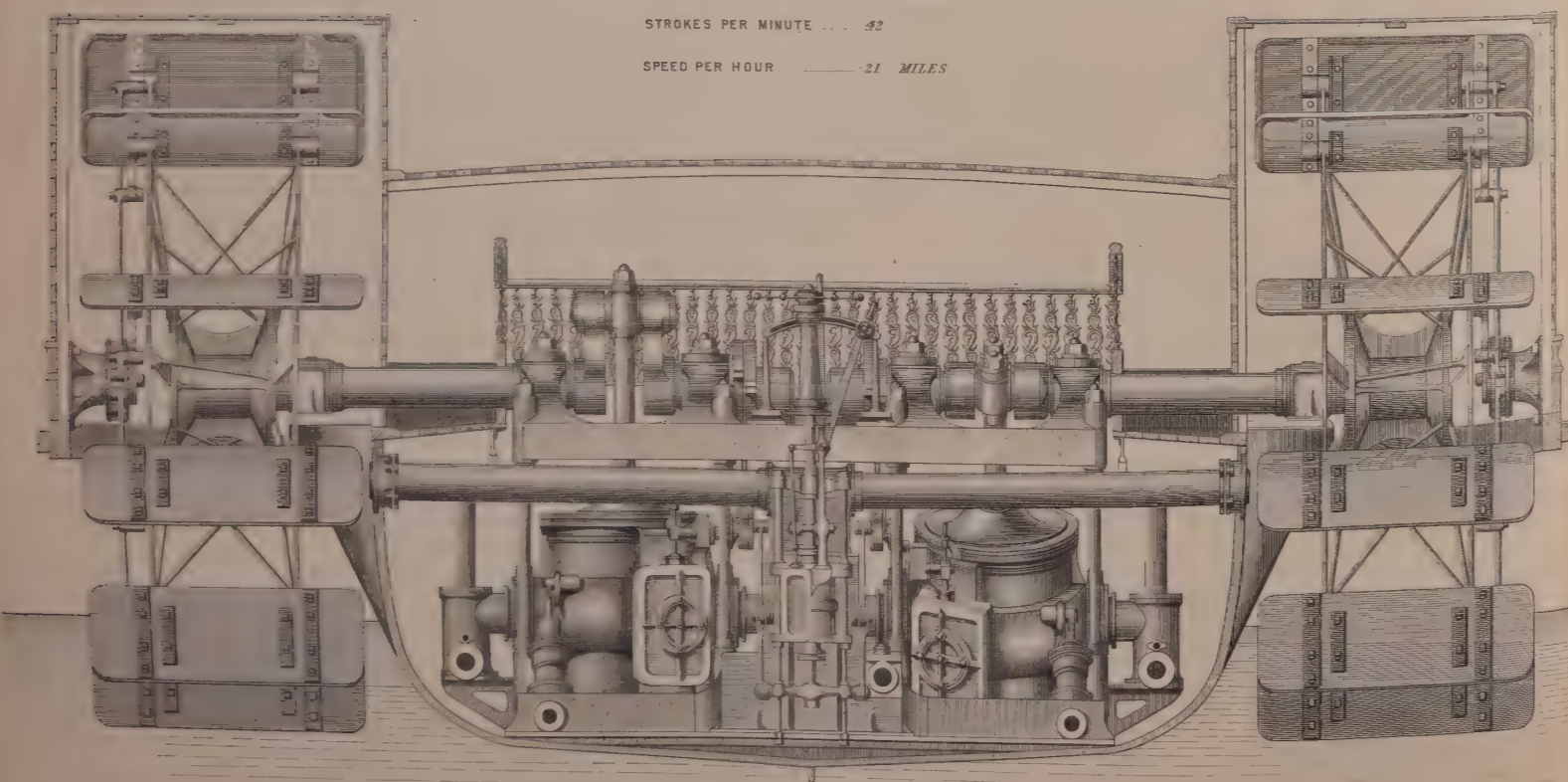
WATERLINE, GUN DECK, AND MAIN DECK



CONSTRUCTED BY JAMES AND GEORGE THOMSON, GLASGOW

PRINCIPAL PARTICULARS

DIAMETER OF CYLINDER $50\frac{1}{2}$ INCHES
 LENGTH OF STROKE 51 INCHES
 STROKES PER MINUTE 42
 SPEED PER HOUR 21 MILES



STEEL SAILING SHIP "FORMER"

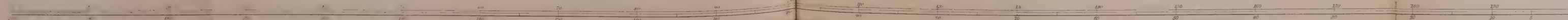
PLATE F

BUILT BY MESSRS JONES, QUINN & CO LIVERPOOL.

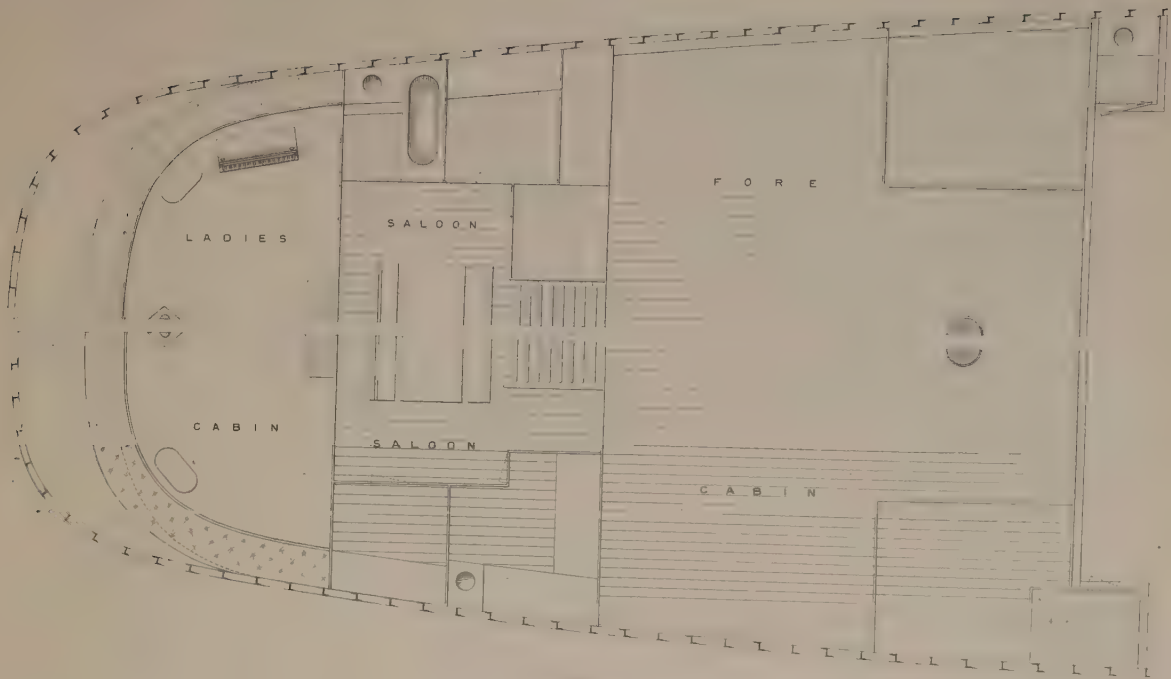
SECTION OF INBOARD WORKS.

DECK PLAN

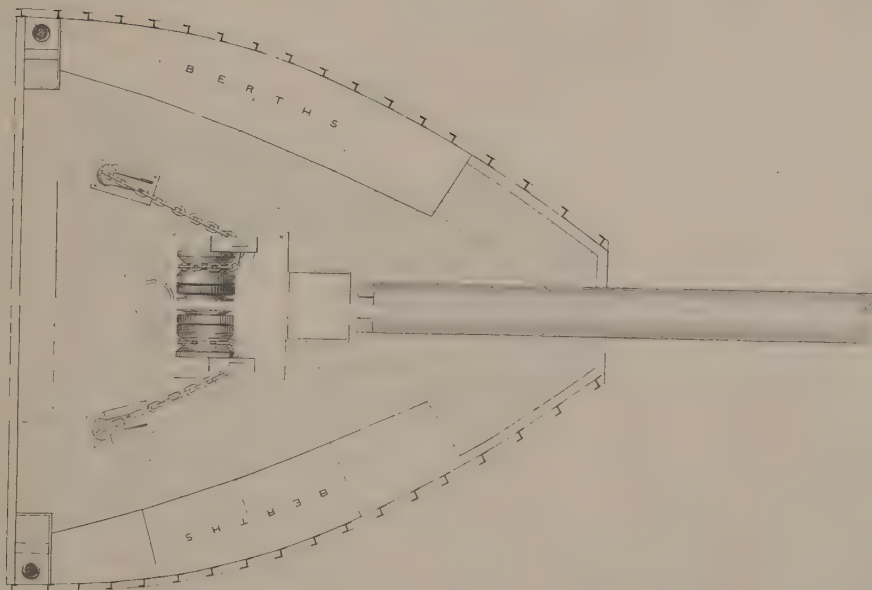
SCALE OF FEET



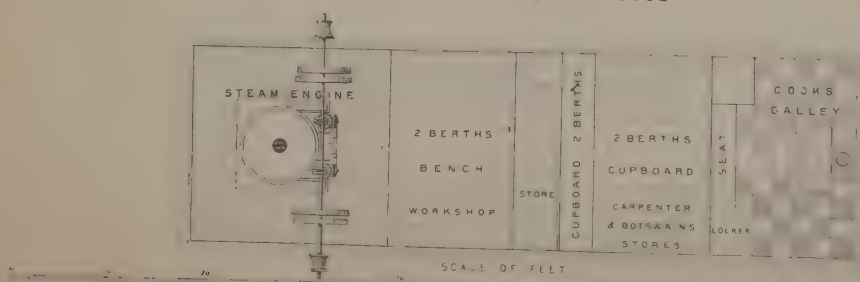
PLAN OF CABINS UNDER POOP



PLAN OF FORECASTLE



PLAN OF DECK-HOUSE



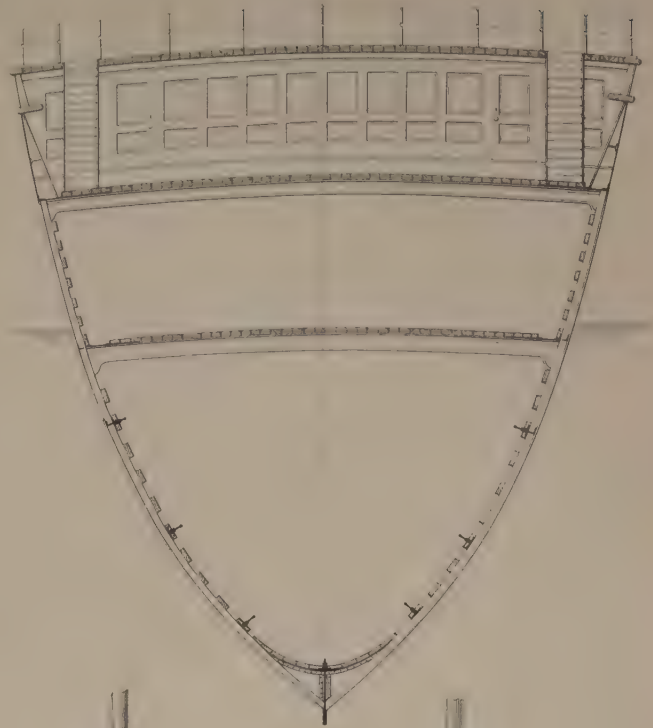
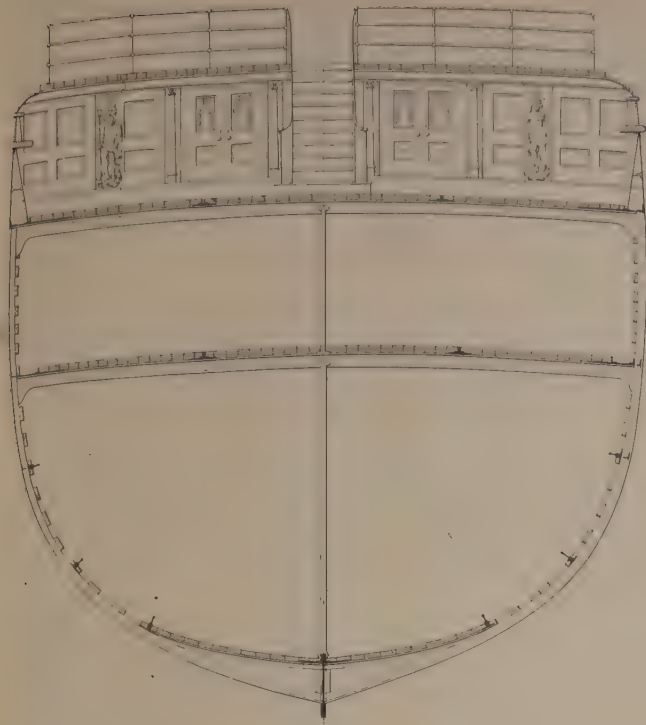
STEEL SAILING SHIP "FORMBY"

PLATE E

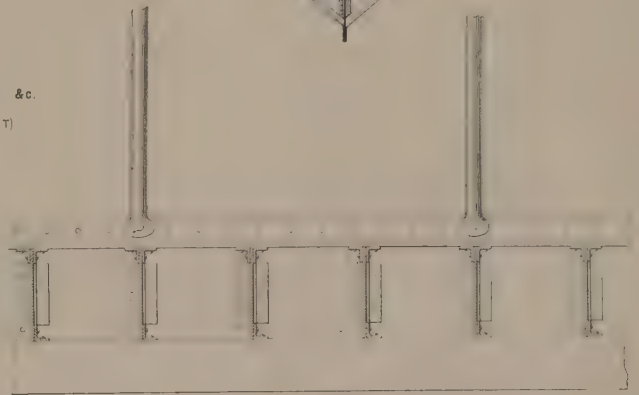
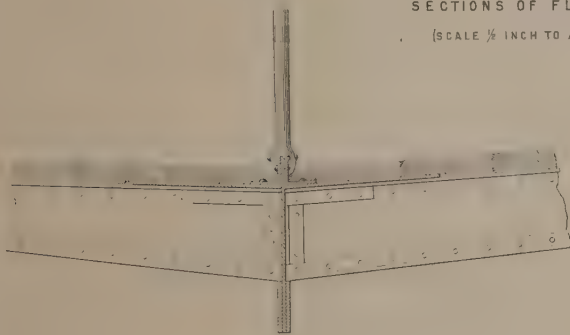
CROSS SECTION AT
D

BUILT BY MESSRS JONES, QUIGGIN, & CO LIVERPOOL.

CROSS SECTION AT
B



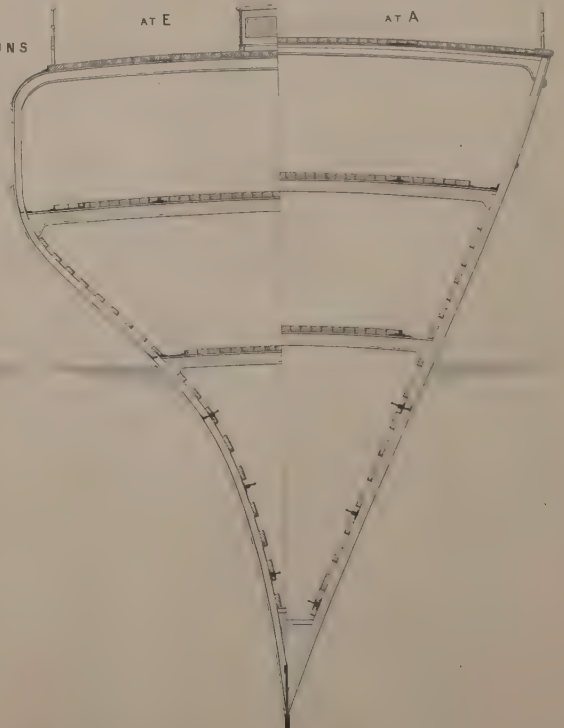
SECTIONS OF FLOOR &c.
(SCALE 1/4 INCH TO A FOOT)



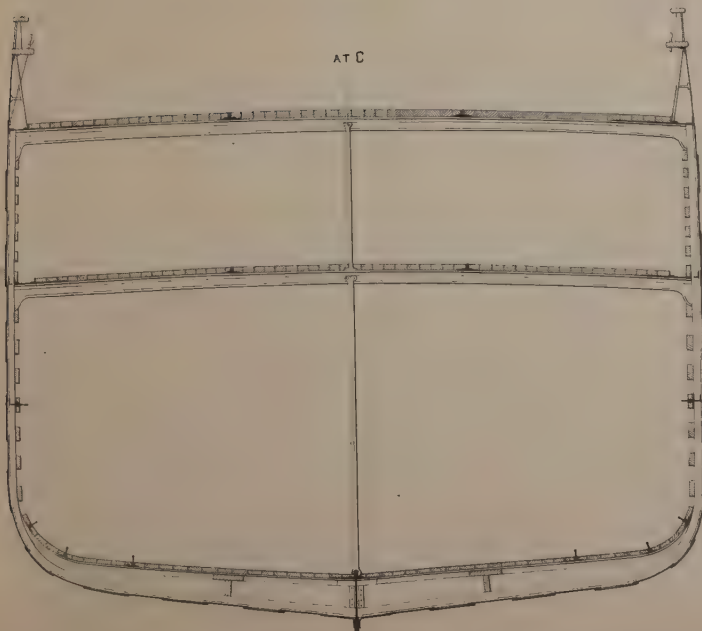
CROSS SECTIONS

AT E

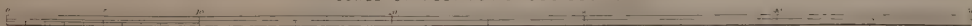
AT A



AT C



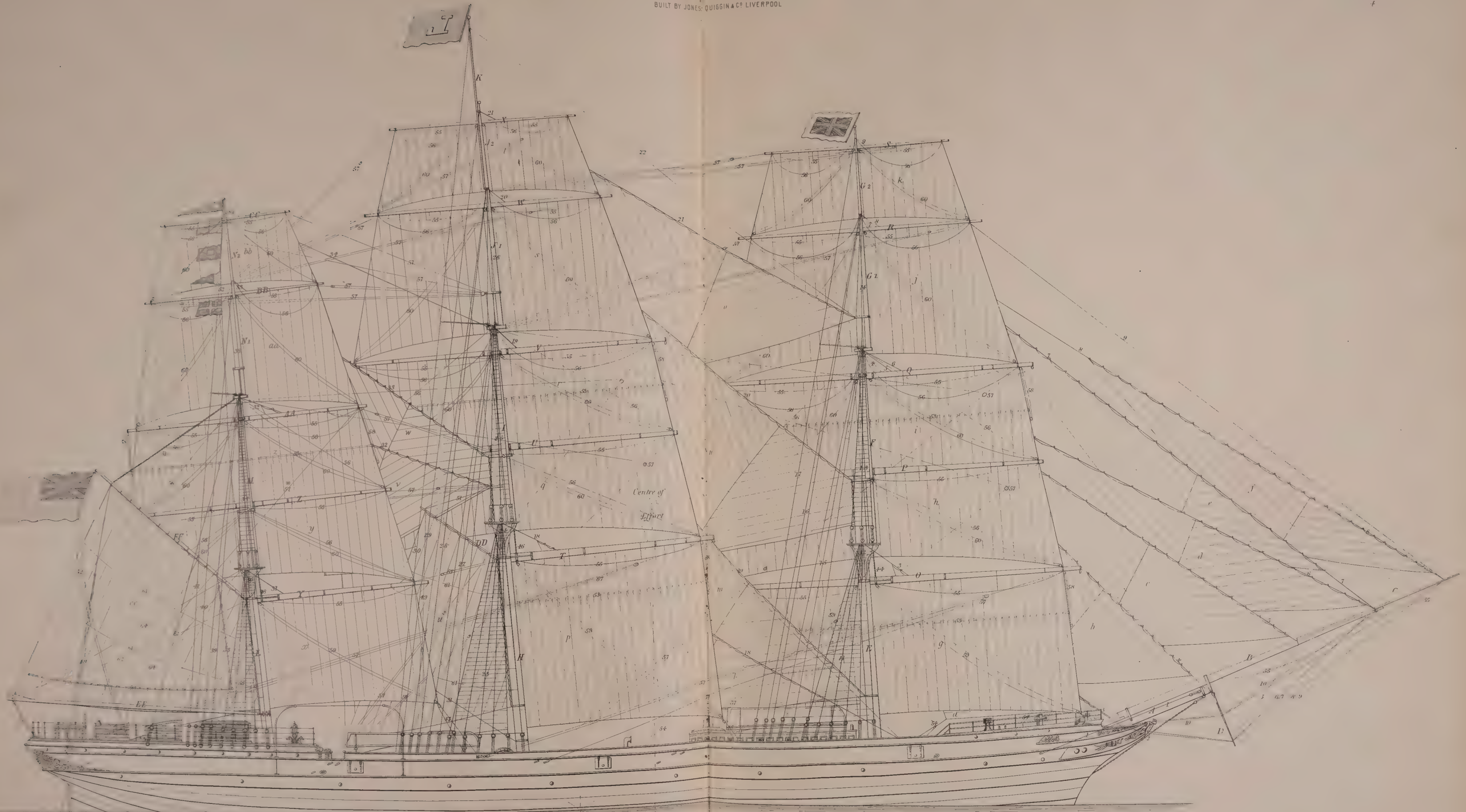
SCALE OF FEET FOR CROSS SECTIONS



RIGGING PLAN OF STEEL SAILING SHIP 'FORMBY'

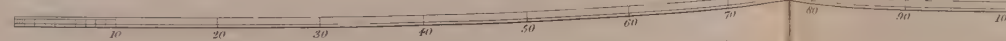
BUILT BY JONES, QUIGGIN & CO LIVERPOOL

PLATE F
4



Centre of Line
of Flotation

Scale of Feet

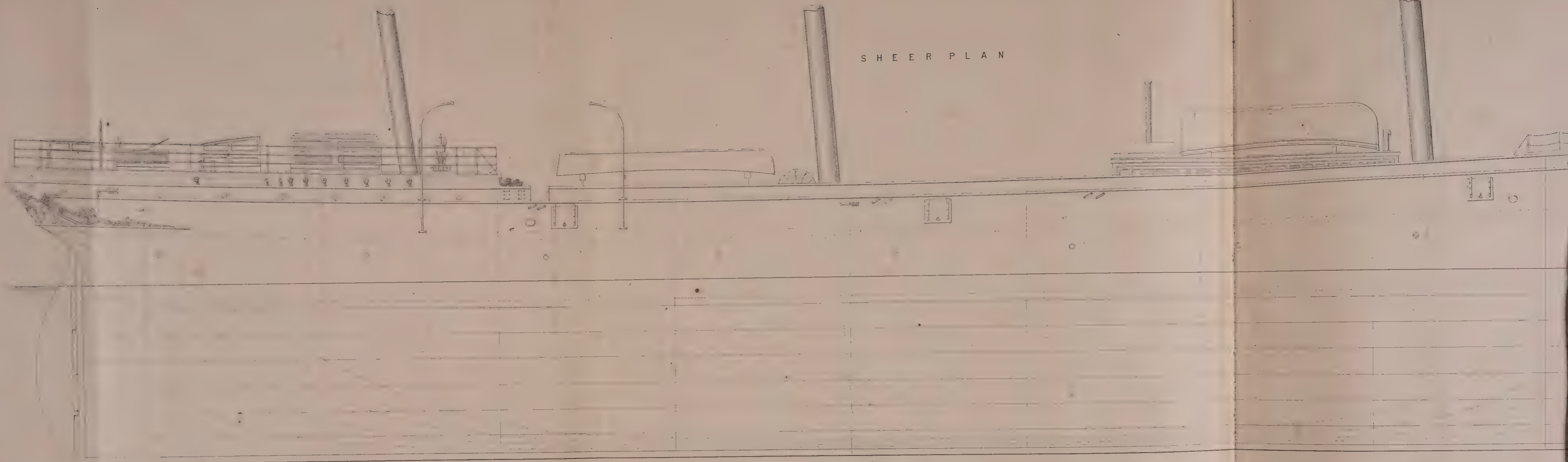


WILLIAM MACKENZIE, GLASGOW, EDINBURGH & LONDON

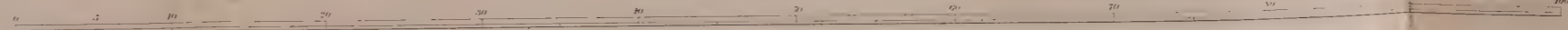
STEEL SAILING SHIP "FORMBY"

BUILT BY MESSRS JONES QUIGGIN & CO. LIVERPOOL

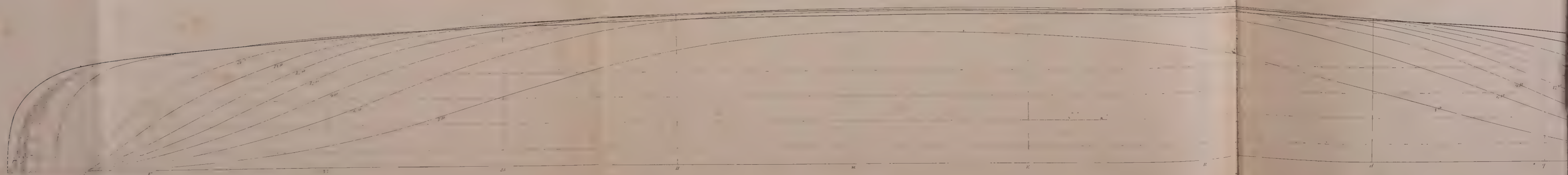
SHEER PLAN



SCALE OF FEET



HALF BREADTH PLAN

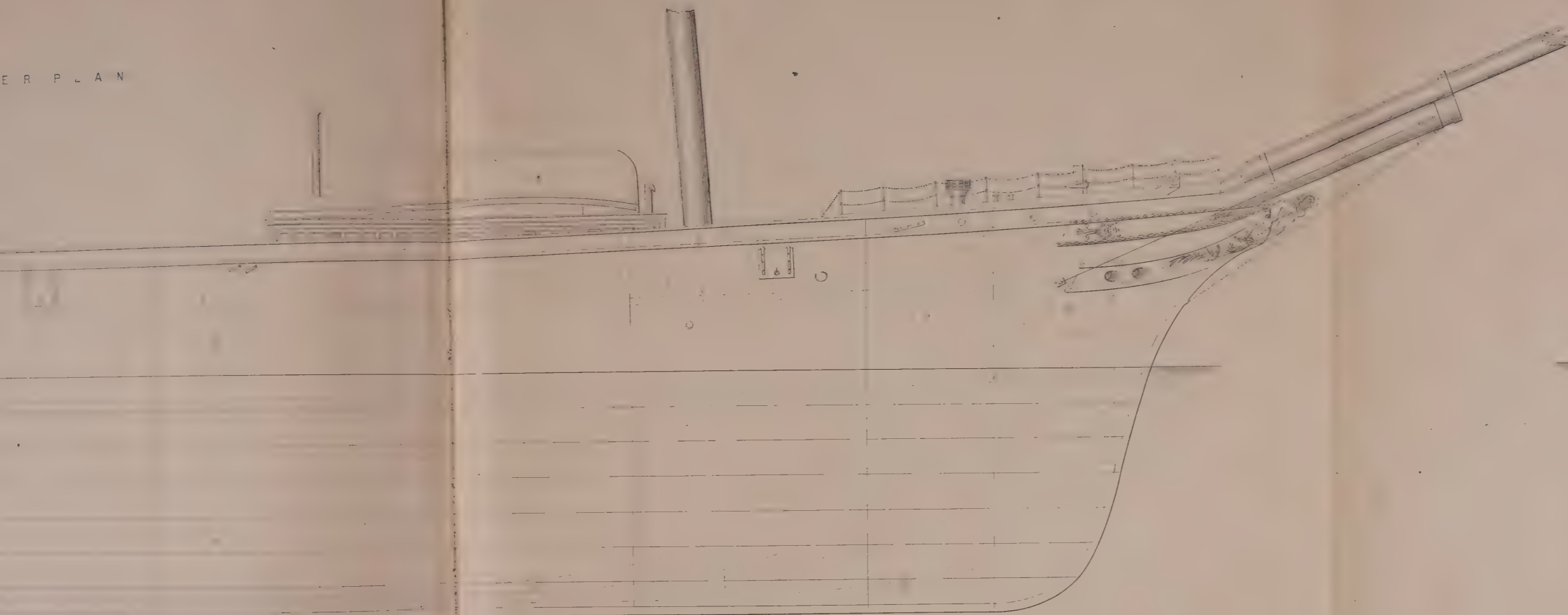


STEEL SAILING SHIP "FORMBY"

BUILT BY MESSRS JONES QUIGG & CO LIVERPOOL

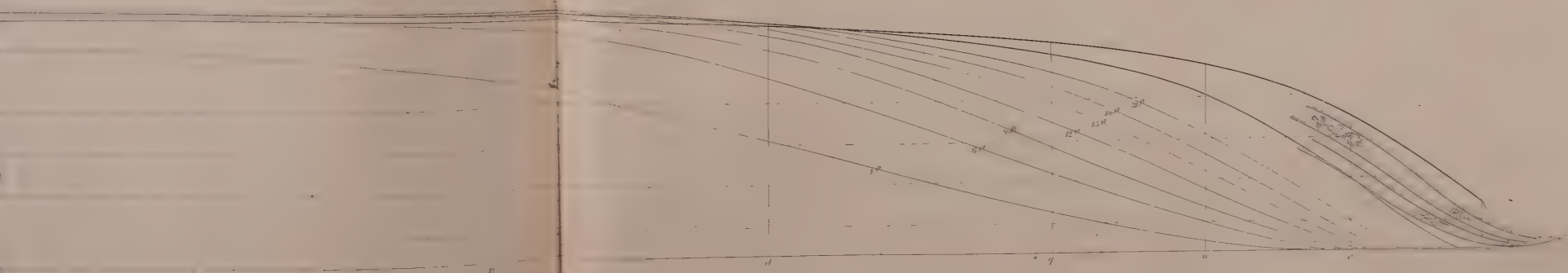
PLATE F

ER PLAN

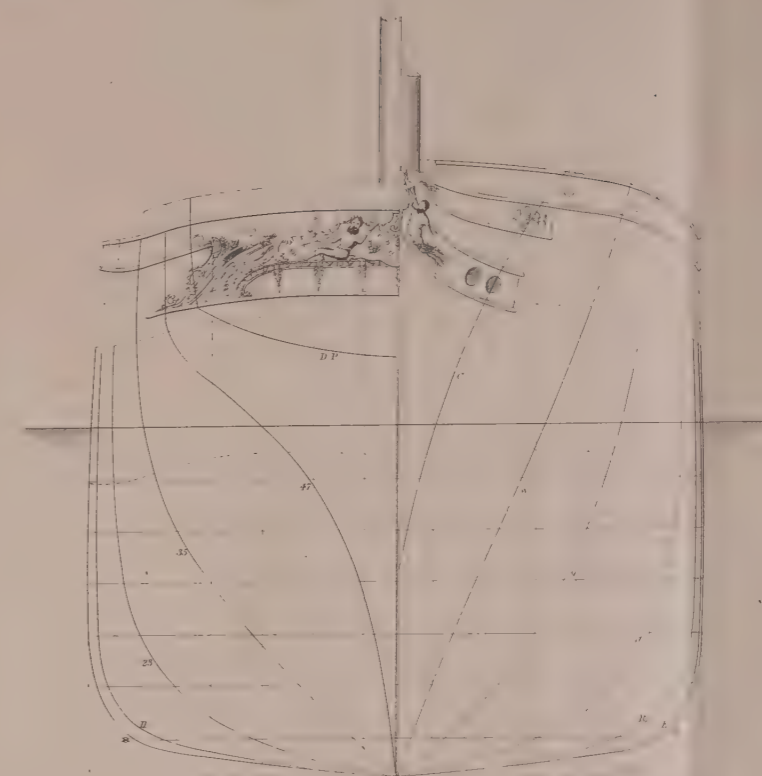


FEET

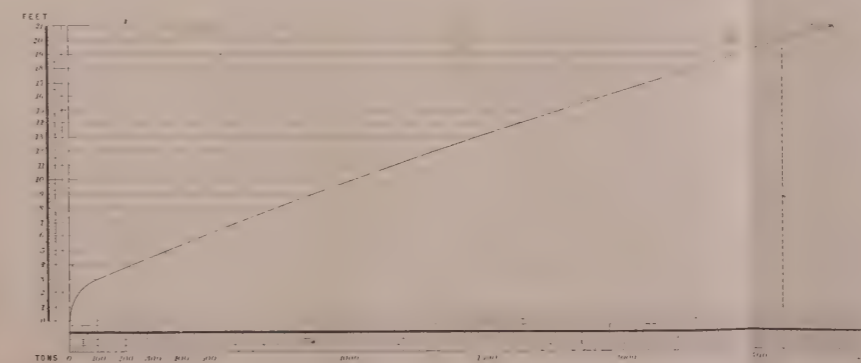
ADTH PLAN



BODY PLAN

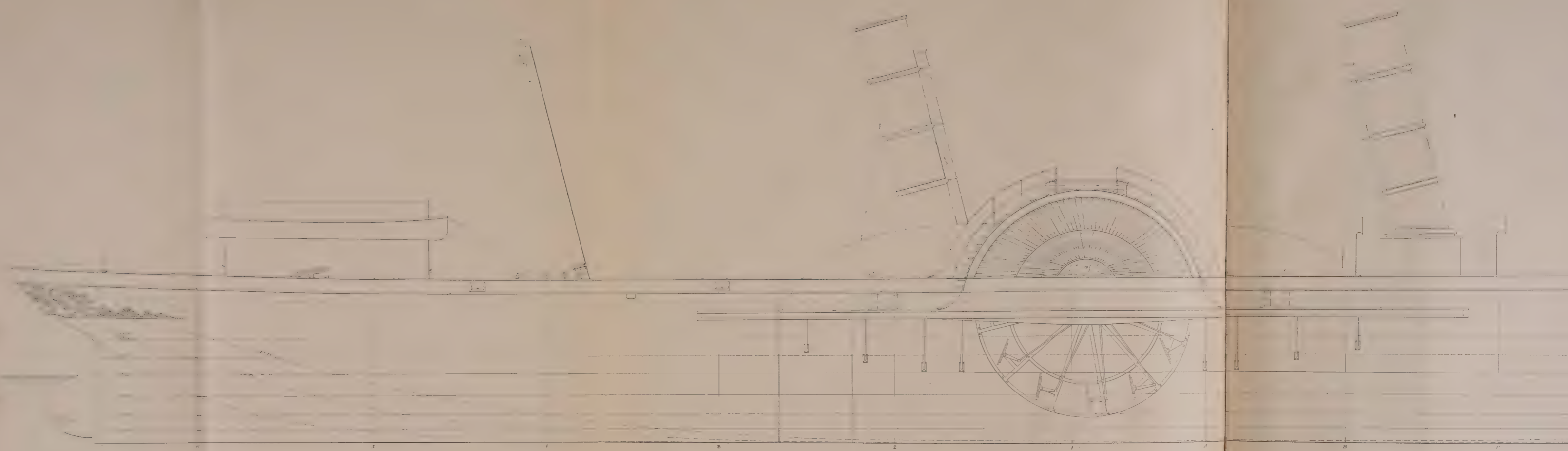


SCALE OF DISPLACEMENT



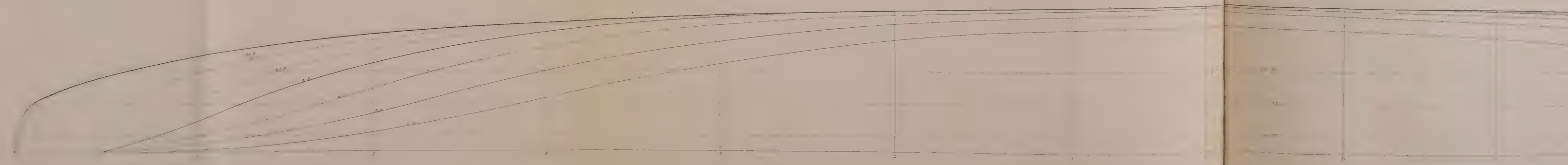
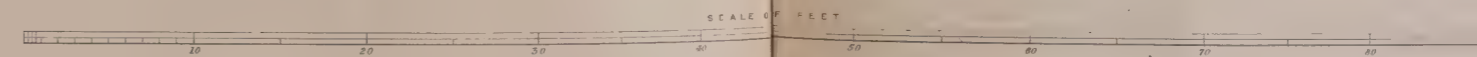
PLAN OF THE HULL OF A STEEL PADDLE STEAMER

BUILT BY MESSRS JONES & CO LIVERPOOL 1864

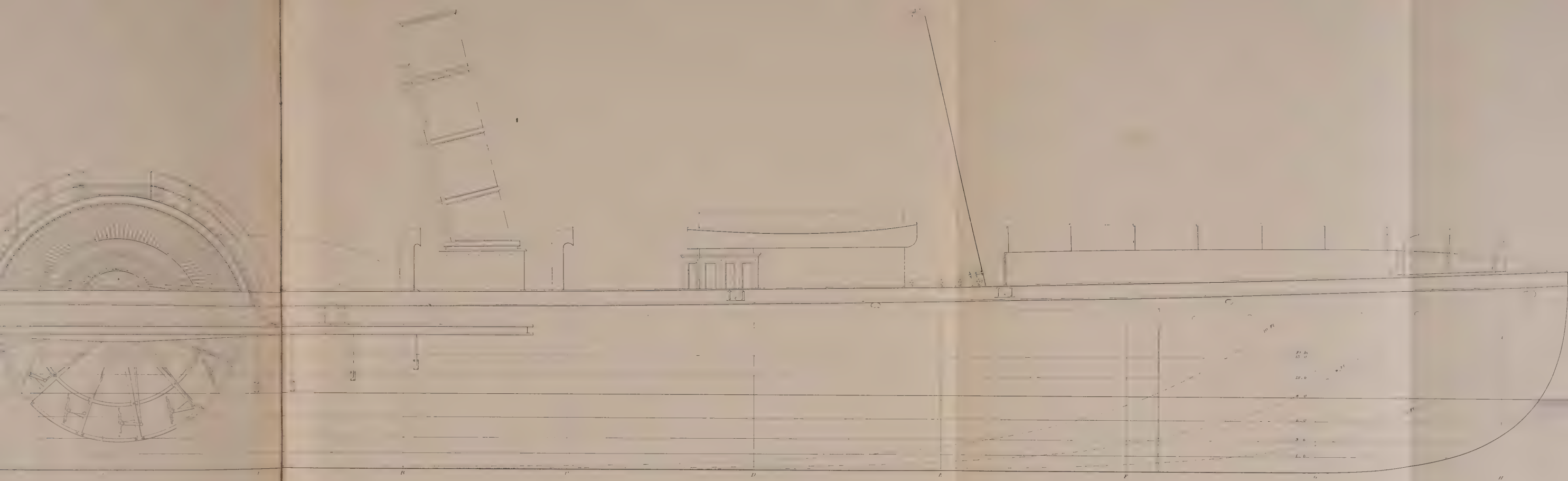


PRINCIPAL DIMENSIONS

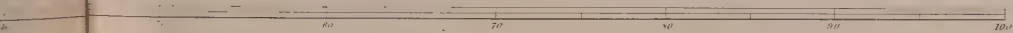
LENGTH BETWEEN PER'S	281' 0"
BREADTH	35' 0"
DEPTH	15' 0"



WILLIAM MACKENZIE, DESIGNER

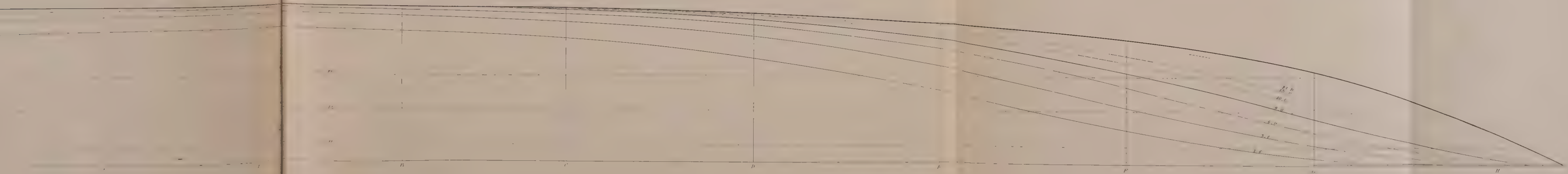


SCALE OF FEET



PARTICULARS

TONNAGE O M 1698
 POWER OF ENGINES 550 H.P. NOM.
 SPEED PER HOUR 18 MILES

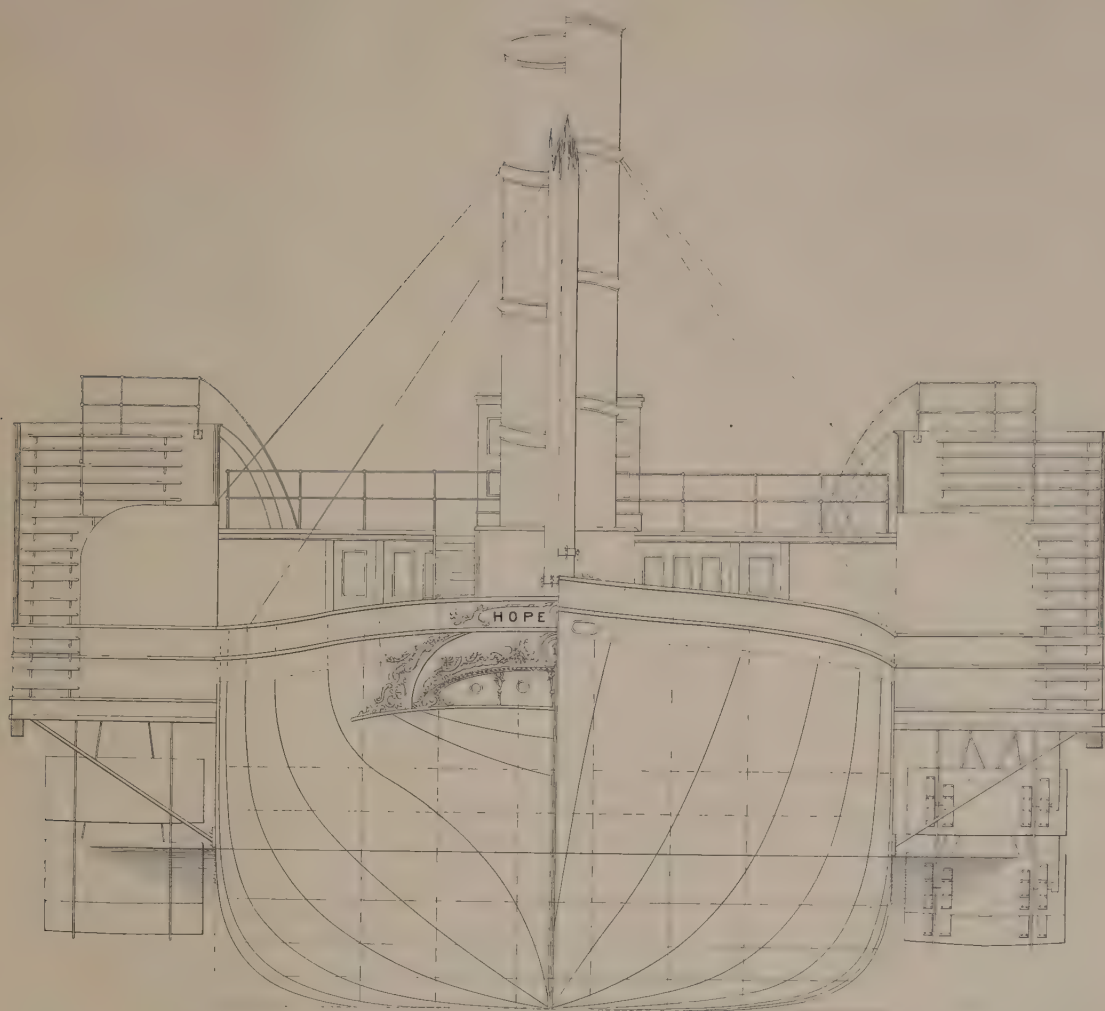


STEEL PADDLE STEAMER 'HOPE'

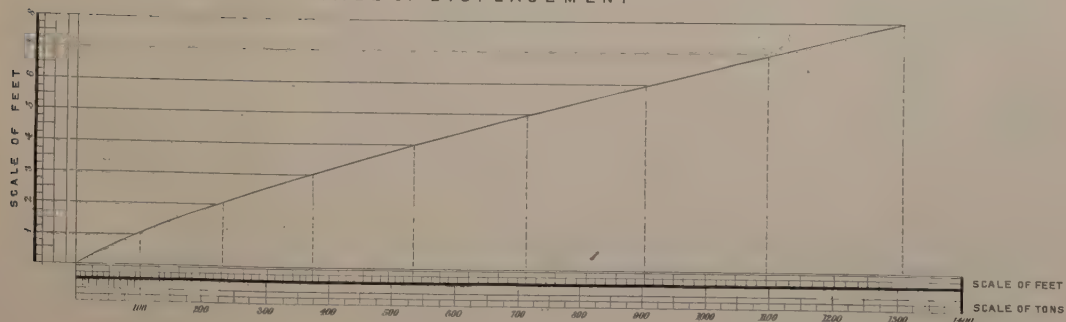
PLATE 6

BUILT BY MESSRS JONES, QUIGGIN & CO LIVERPOOL.

BODY PLAN



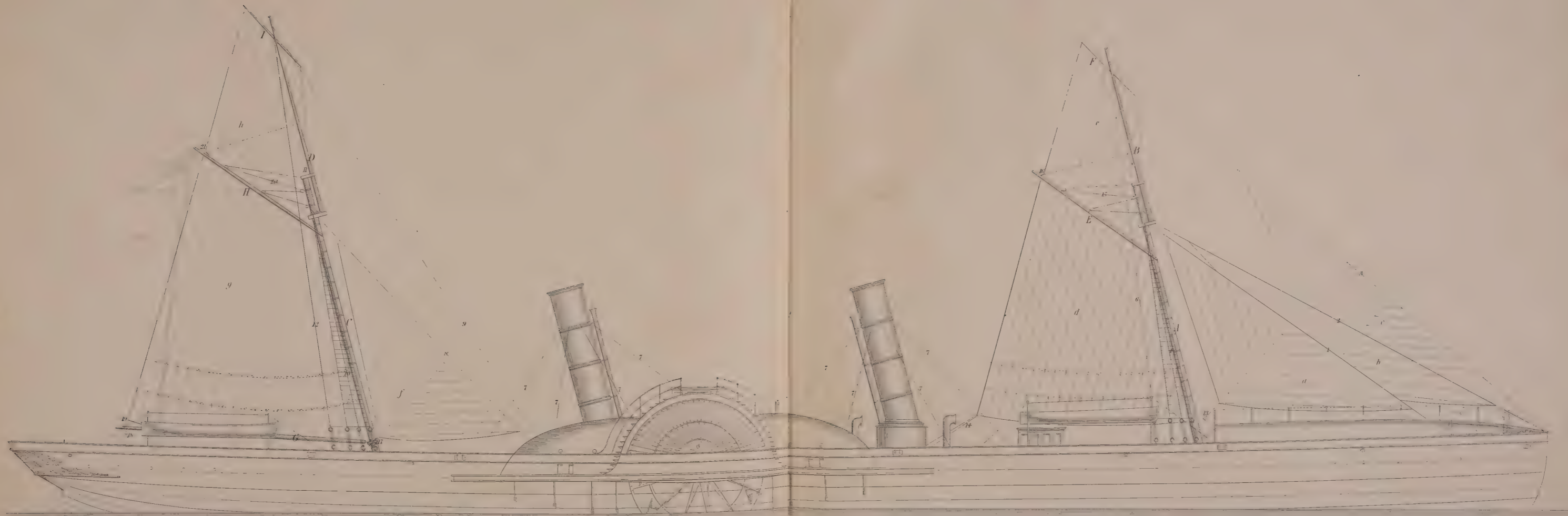
SCALE OF DISPLACEMENT



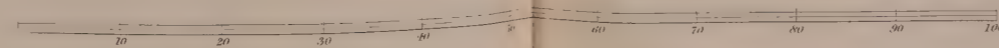
RIIDING PLAN OF STEEL-PLATE STEAMER HOPE

PLATE 6
3

BUILT BY MESSRS JONES QUIGGIN & CO LIVERPOOL



Scale of Feet

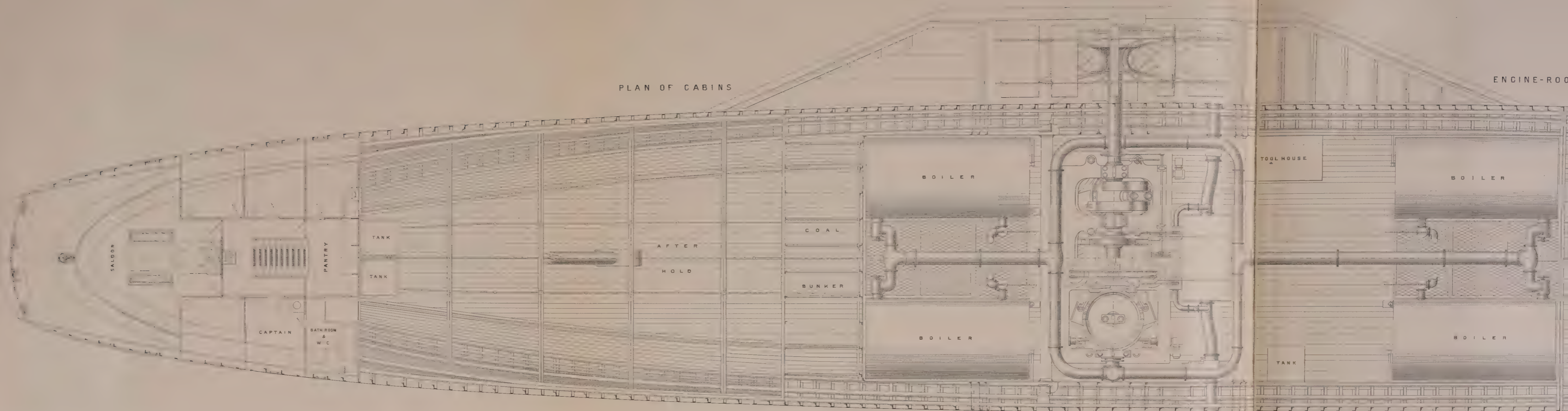


WILLIAM MACKENZIE, LONDON & G. ASGOW.

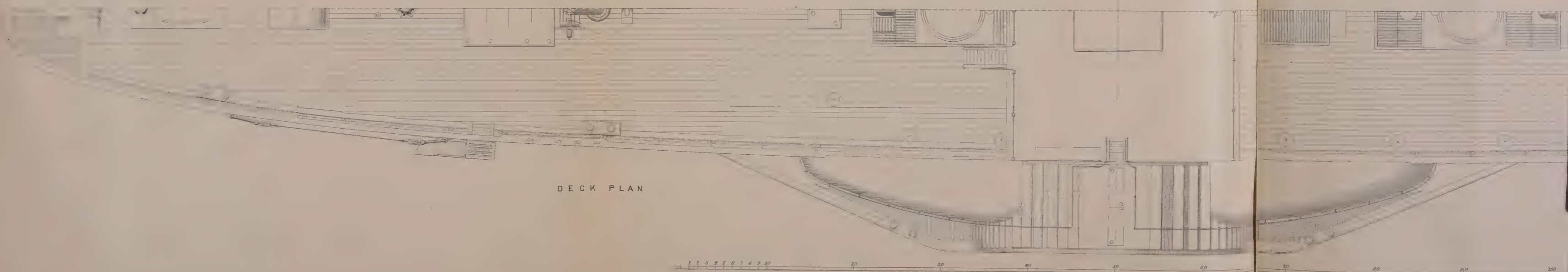
STEEL PADDLE STEAMER 'HOPE'

BUILT BY MESSRS JONES QUIGGIN & CO LIVERPOOL.

PLAN OF CABINS



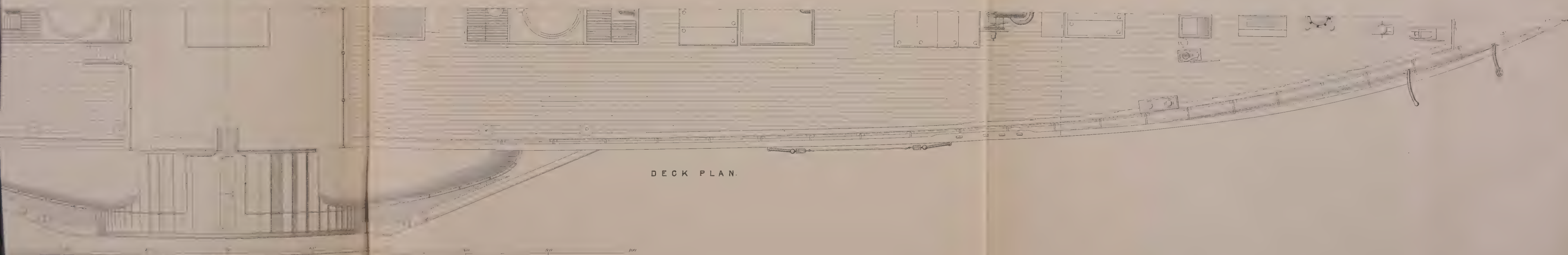
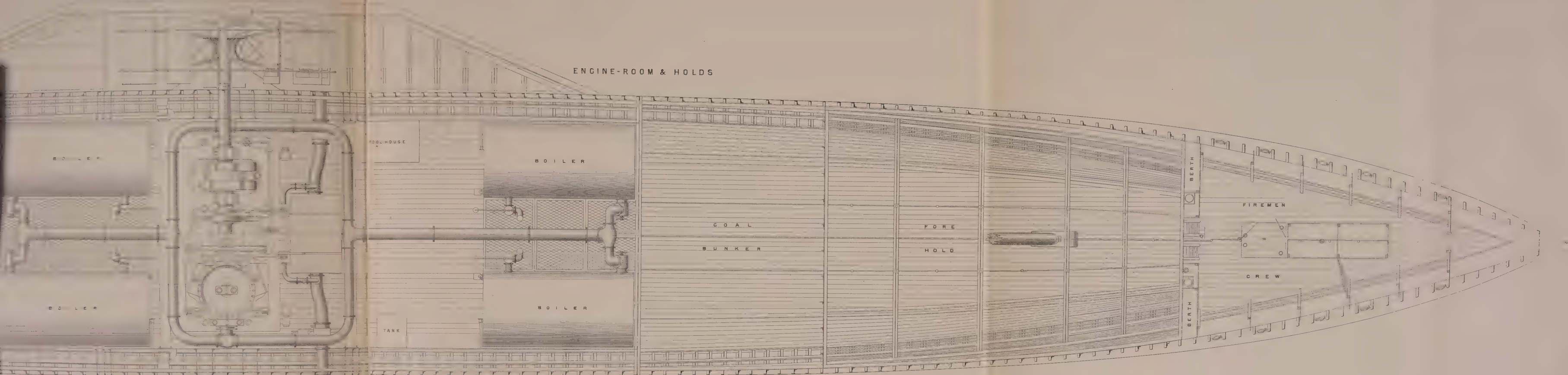
DECK PLAN



STEEL PADDLE STEAMER "MD"

BUILT BY MESSRS JONES QUIGGIN & CO LIVERPOOL.

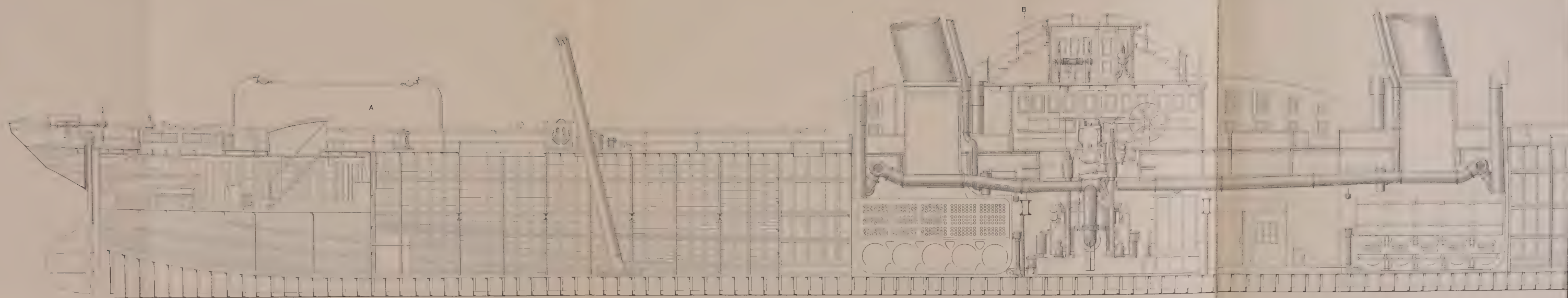
PLATE G



STEEL PADDLE STEAMER 'HOPE'

BUILT BY MESSRS JONES & QUIGGIN & CO LIVERPOOL.

LONGITUDINAL SECTION SHEWING GENERAL ARRANGEMENTS



WILLIAM WALKER & CO. GLASGOW, EDINBURGH & LONDON.

STEEL PADDLE STEAMER HOPE

BUILT BY MESSRS JONES QUIGGIN & CO LIVERPOOL

PLATE G
5

LONGITUDINAL SECTION SHEWING GENERAL ARRANGEMENTS

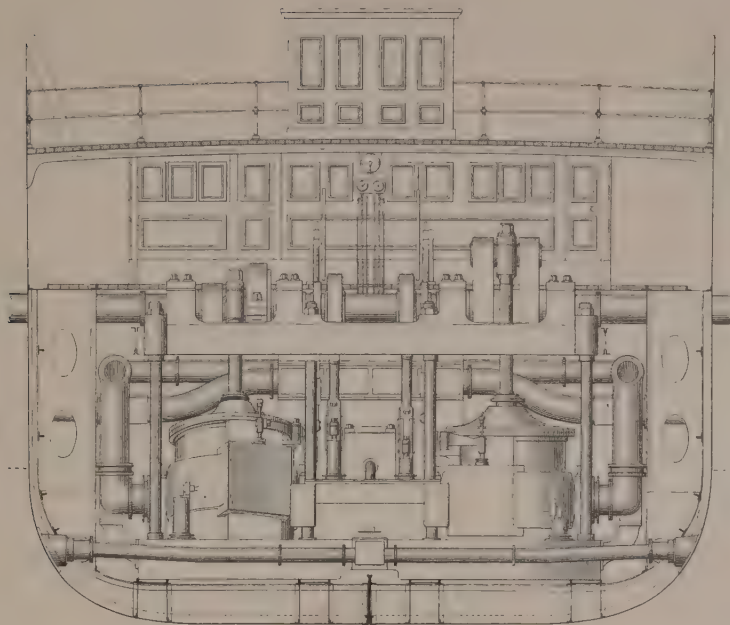
WILLIAM MACKENZIE & CO. ASCOW EDINBURGH & LONDON

STEEL PADDLE STEAMER 'HOPE'

PLATE 6

BUILT BY MESSRS JONES, QUIGGIN & CO LIVERPOOL

SECTION AT B
SHOWING ENGINES.



SECTION AT A

SECTION AT C

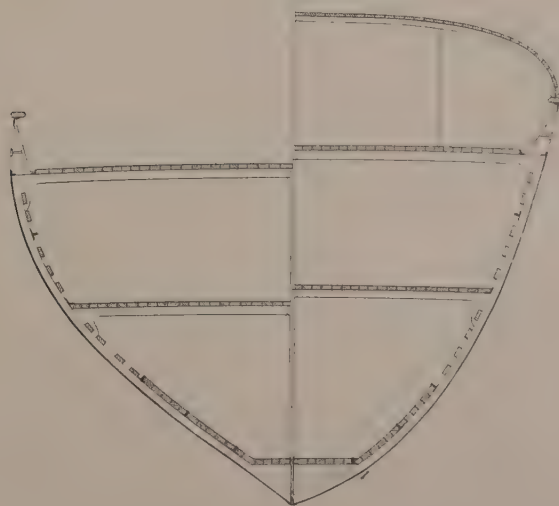


ILLUSTRATION OF RULES FOR BUILDING WOODEN SHIPS.

COPIED FROM LLOYD'S REGISTER BY PERMISSION OF THE COMMITTEE.

FIG. 1. ARRANGEMENT OF THROUGH BOLTS IN THICK STRAKES OVER DOUBLE FLOORS.

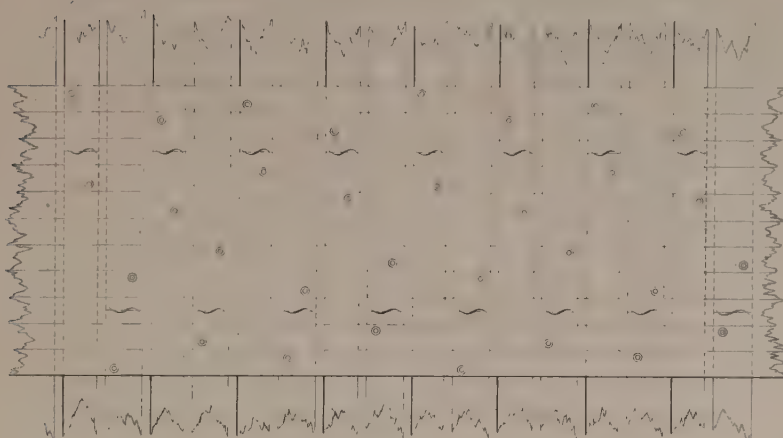
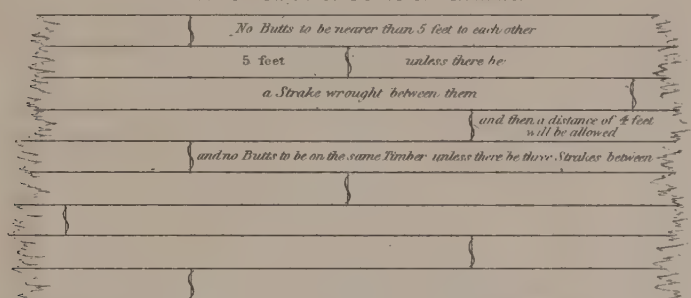


FIG. 3. SHIFT OF BUTTS OF PLANKING.



The sketch shows the principle on which the Butts should be arranged so as to avoid Stepping, which is deemed bad workmanship

FIG. 2. SHEWING THE DIRECTION OF THE IRON PLATES ON FRAMES AND IRON KNEES AND RIDERS.

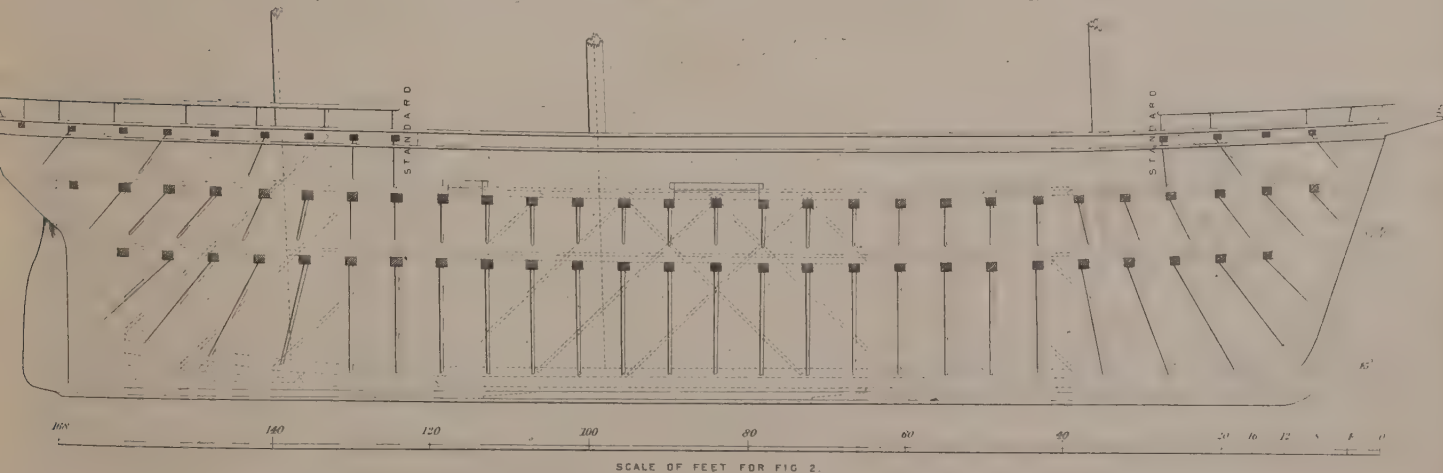
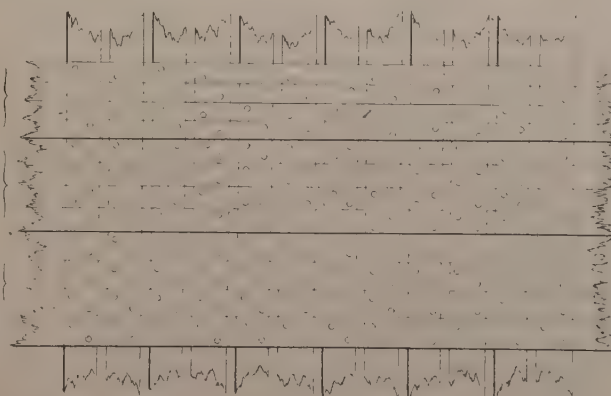


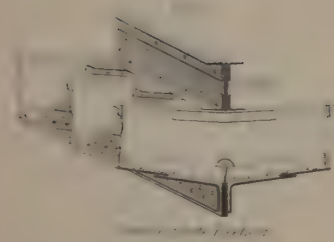
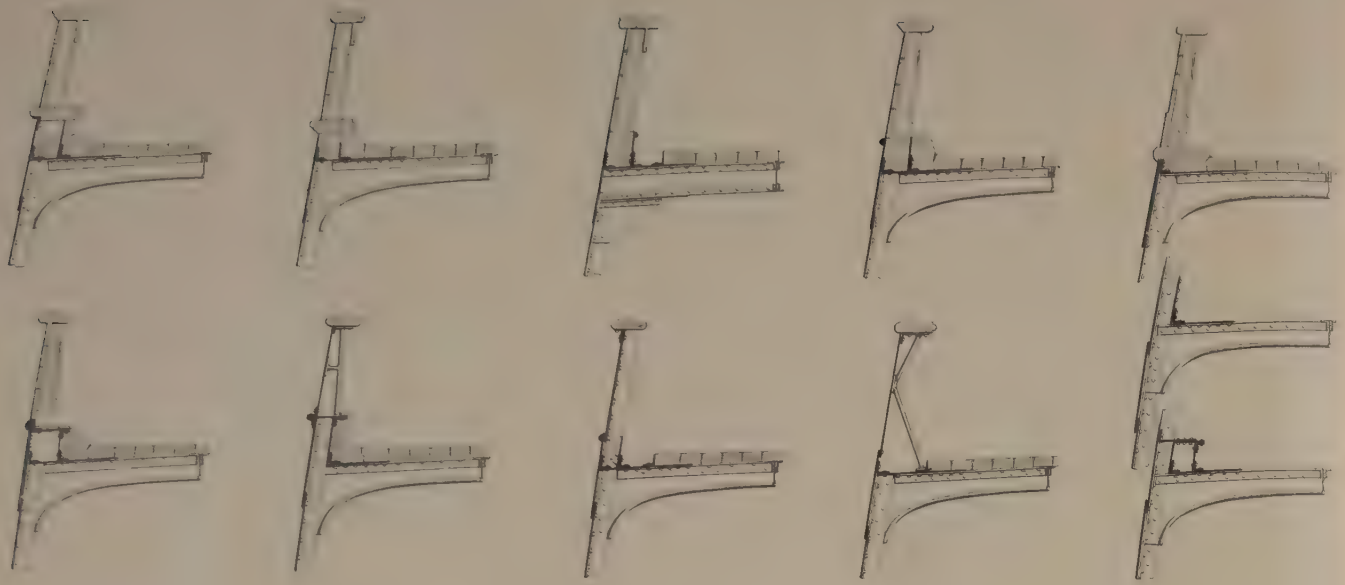
FIG. 4. ARRANGEMENT OF TREENAILS OR BOLTS.

SINGLE FASTENING
 IN PLANKS 8 INCHES WIDE AND UNDER

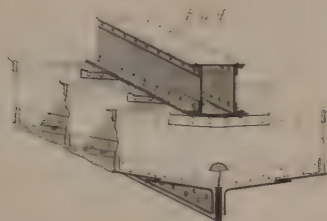
DOUBLE & SINGLE
 FASTENING IN PLANKS ABOVE 8 INCHES
 AND NOT ABOVE 11 INCHES.

DOUBLE FASTENING
 IN PLANKS ABOVE 11 INCHES

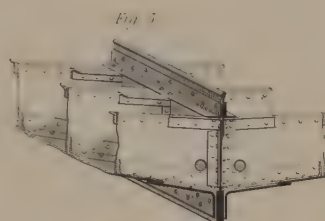




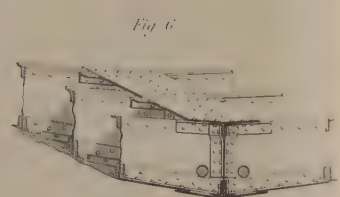
Keel and keelson



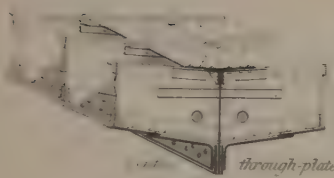
Low keelson



Intervall middle line keelson

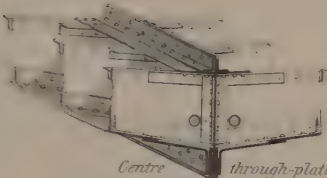


*Flat plate keel and
 Centre and flat plate keelson*



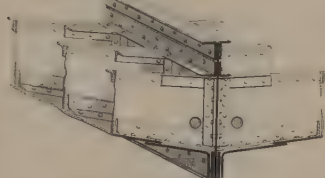
*through-plate
 and flat keelson plate*

Fig. 7.



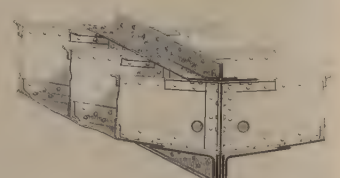
*Centre through-plate
 and two flat keelson plates*

Fig. 8.



Centre through-plate keelson

Fig. 9.



Centre through-plate keelson

Fig. 10.

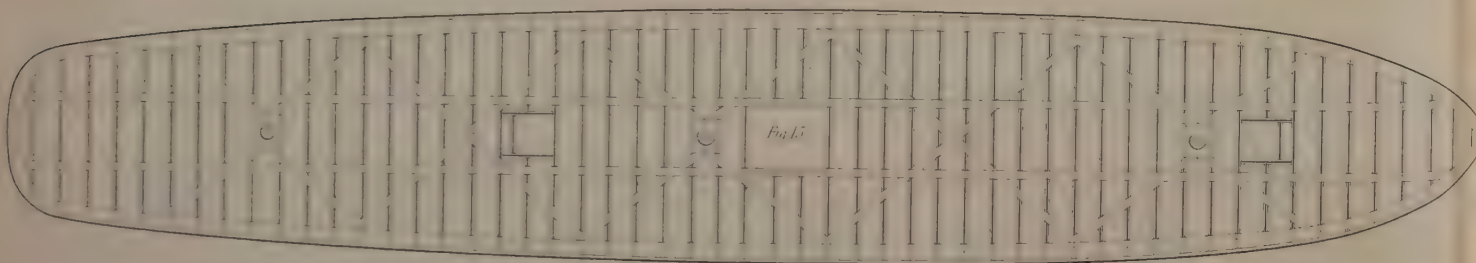


Fig. 12.

Keel and keelson to be riveted together all over and up upon each bar of beams on each side of bulwarks and wherever practicable from side to side, diagonally.

Fig 2

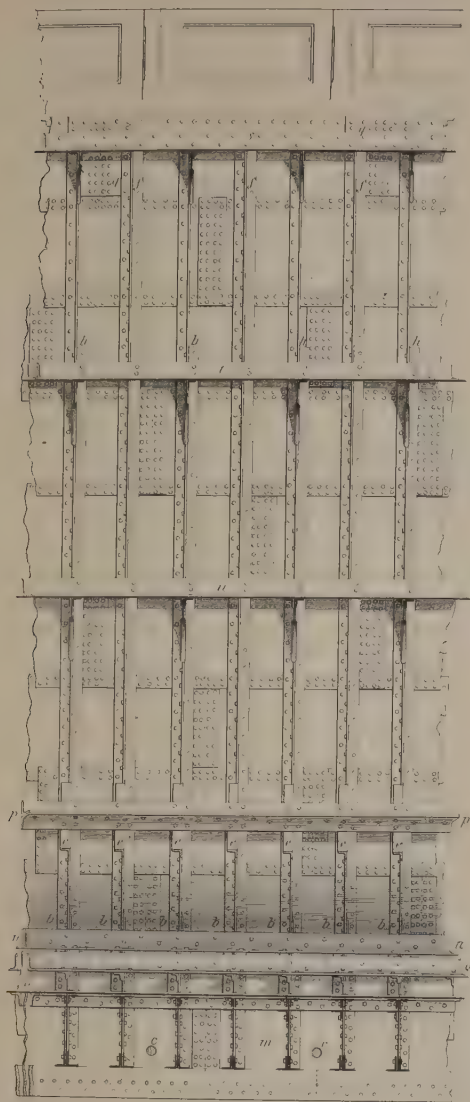
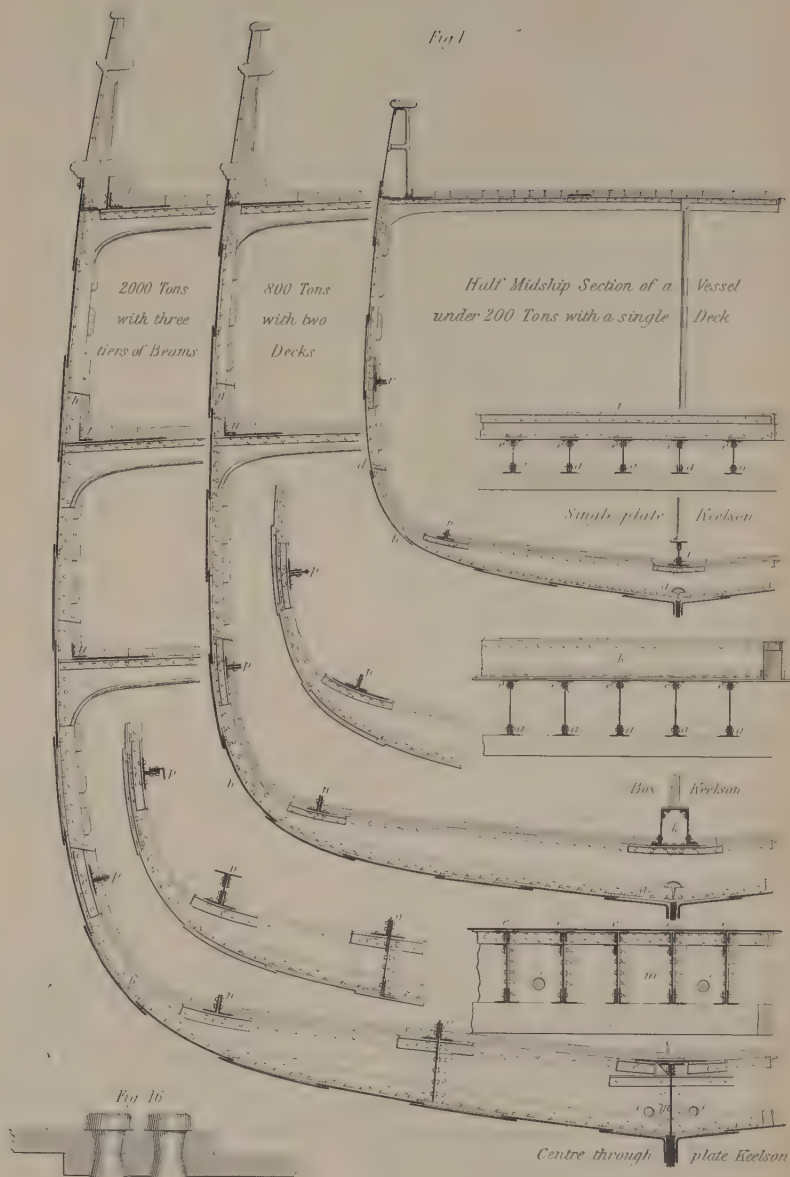
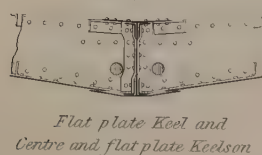


Fig 1



The rivet holes to be punched from the facing surfaces and to be countersunk all through the outer Plating

Fig 11



Flat plate Keel and
Centre and flat plate Keelson

Fig 12



Intercostal middle line Keelson

Fig 13

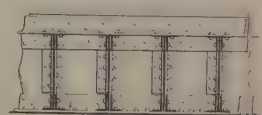
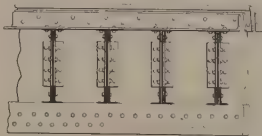


Centre through-plate Keelson

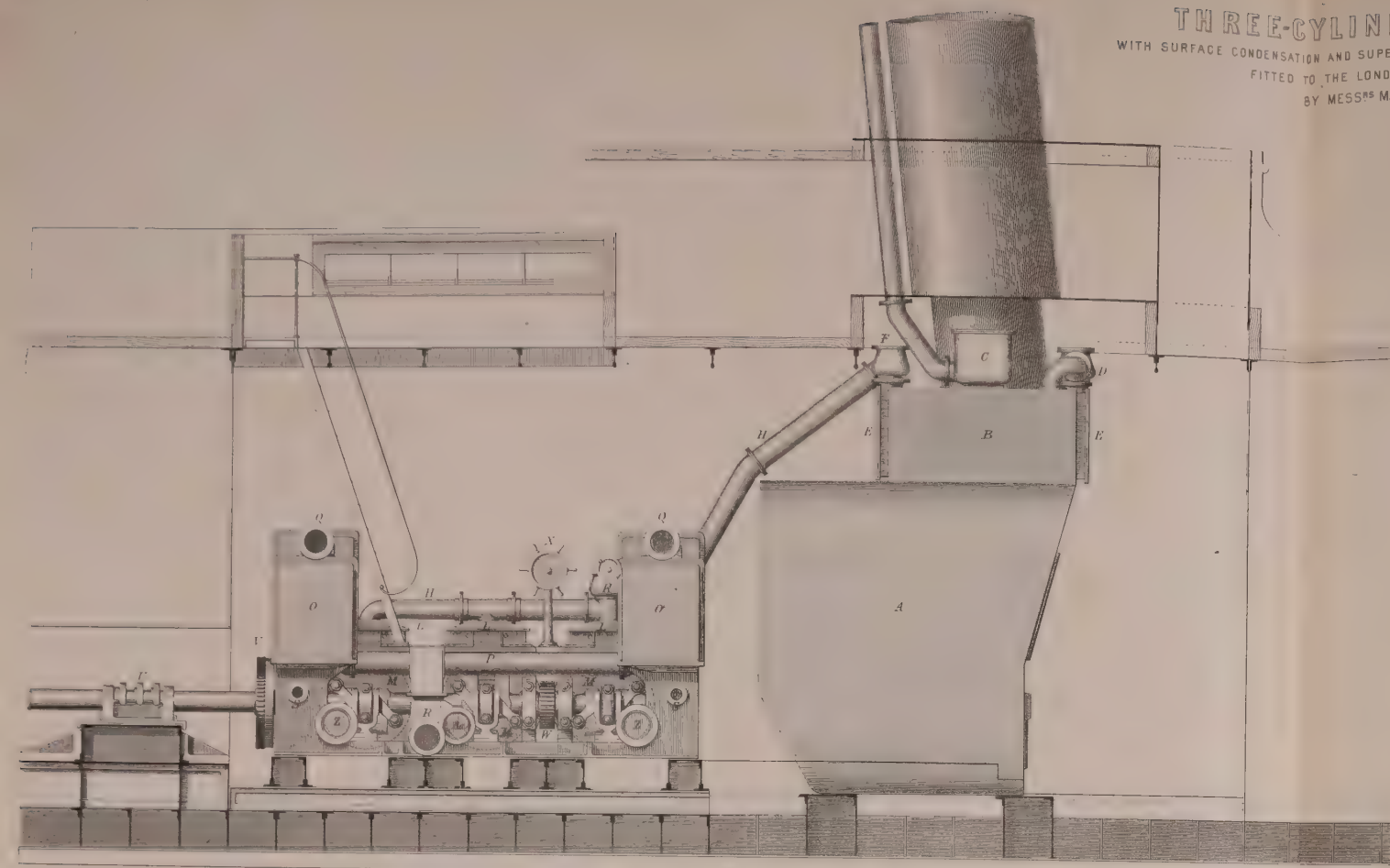
Fig 14



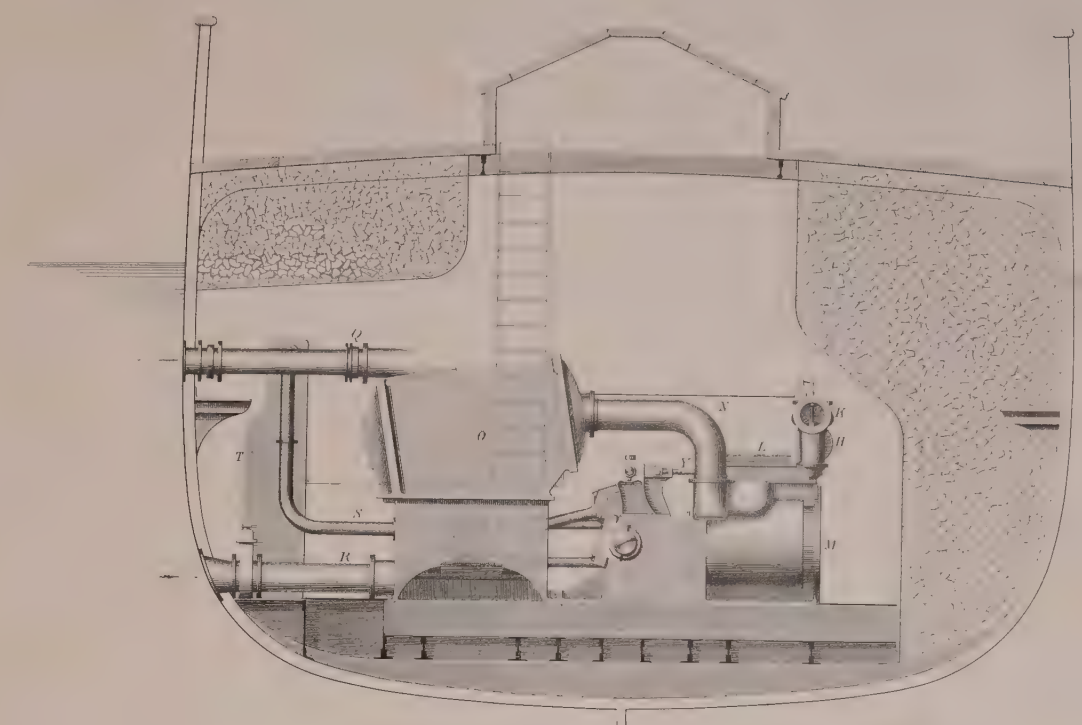
Flat plate Keel and
Intercostal middle line Keelson



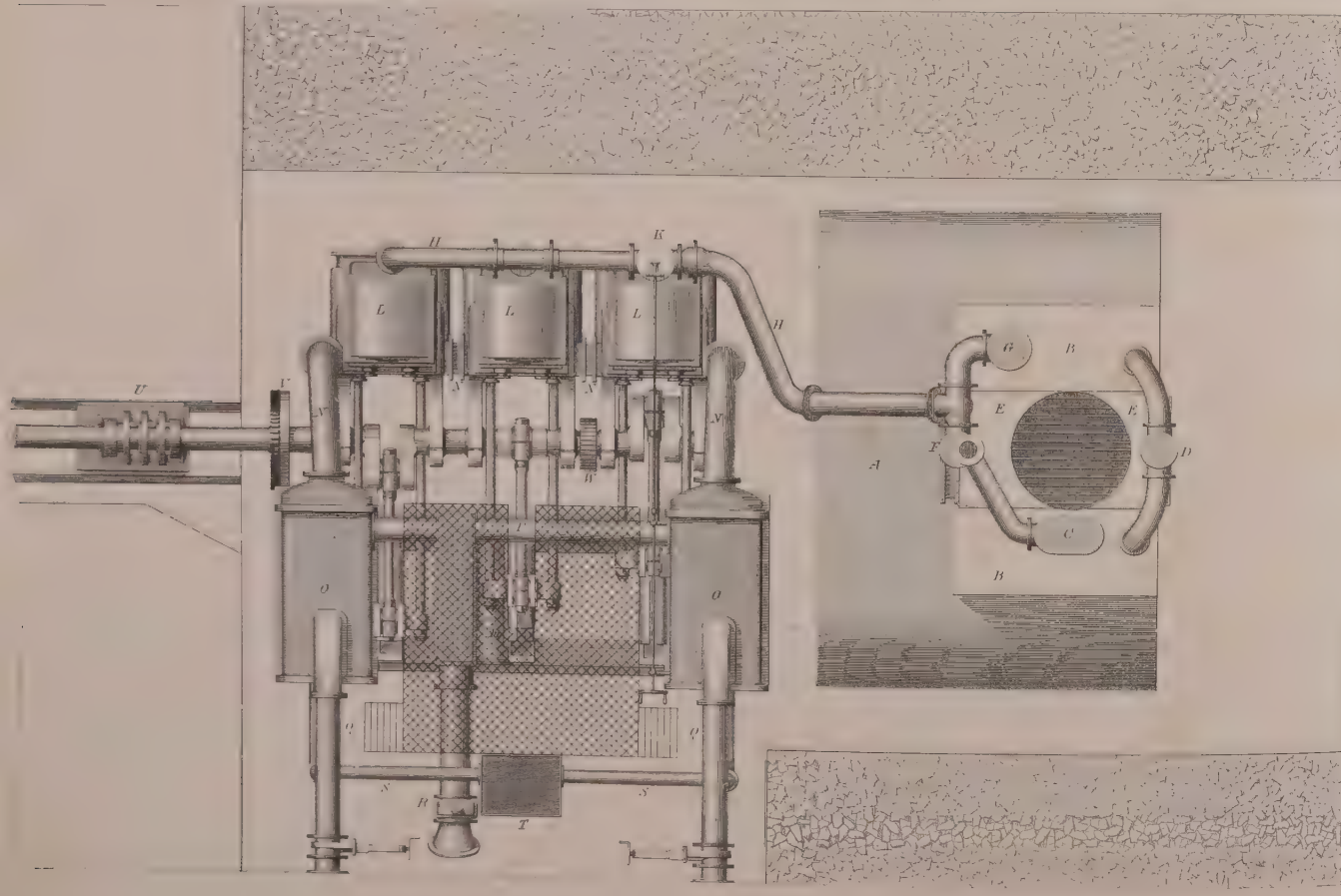
THREE-CYLINDER EXPANSIVE ENGINES,
WITH SURFACE CONDENSATION AND SUPERHEATING APPARATUS OF THE COLLECTIVE POWER OF 150 HORSES NOMINAL
FITTED TO THE LONDON ITALIAN & ADRIATIC SCREW STEAM SHIP (VENETIA)
BY MESS^{RS} MAUDSLAY, SONS & FIELD, ENGINEERS, LONDON.



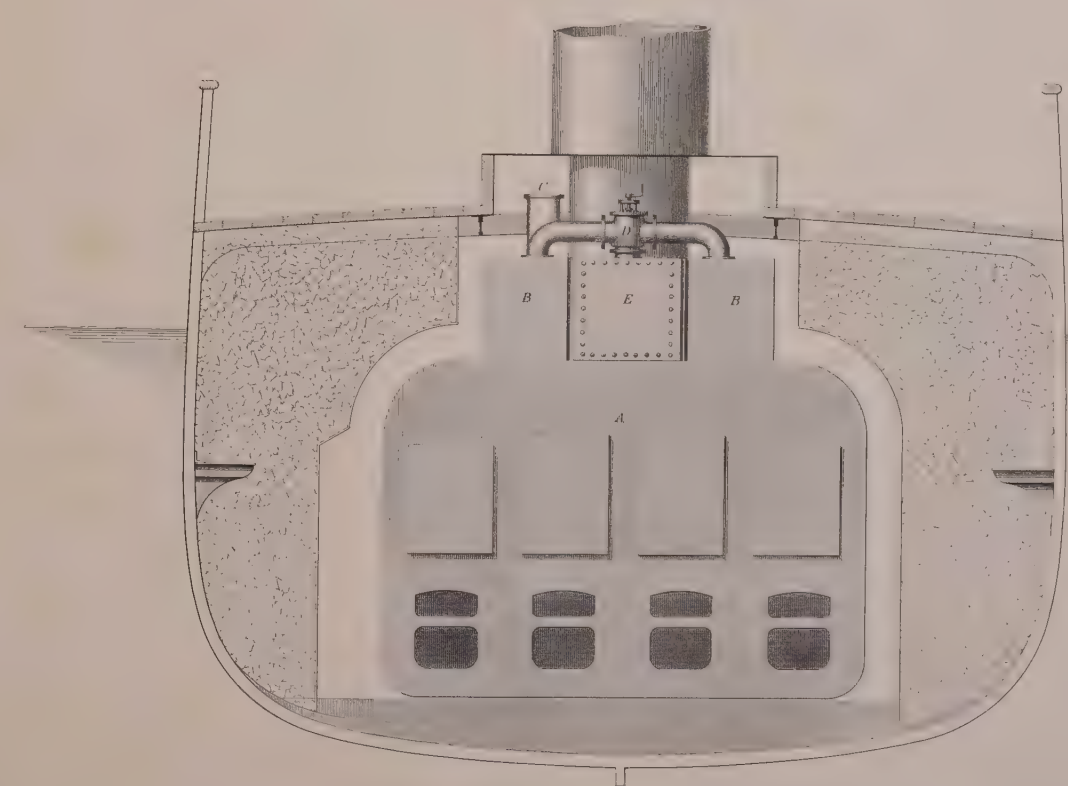
LONGITUDINAL SECTION



SECTION AT ENGINES



PLAN

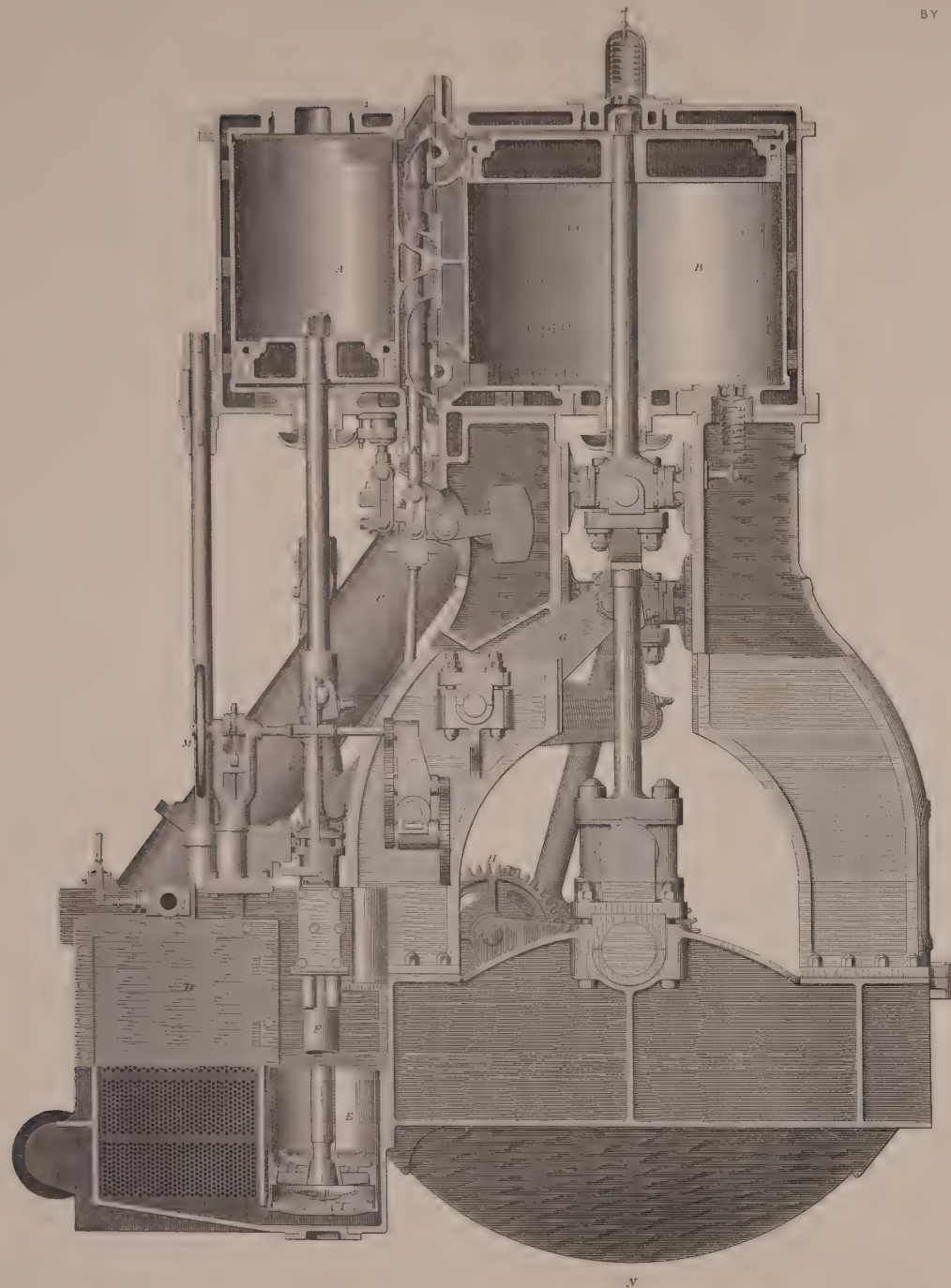


SECTION AT BOILERS

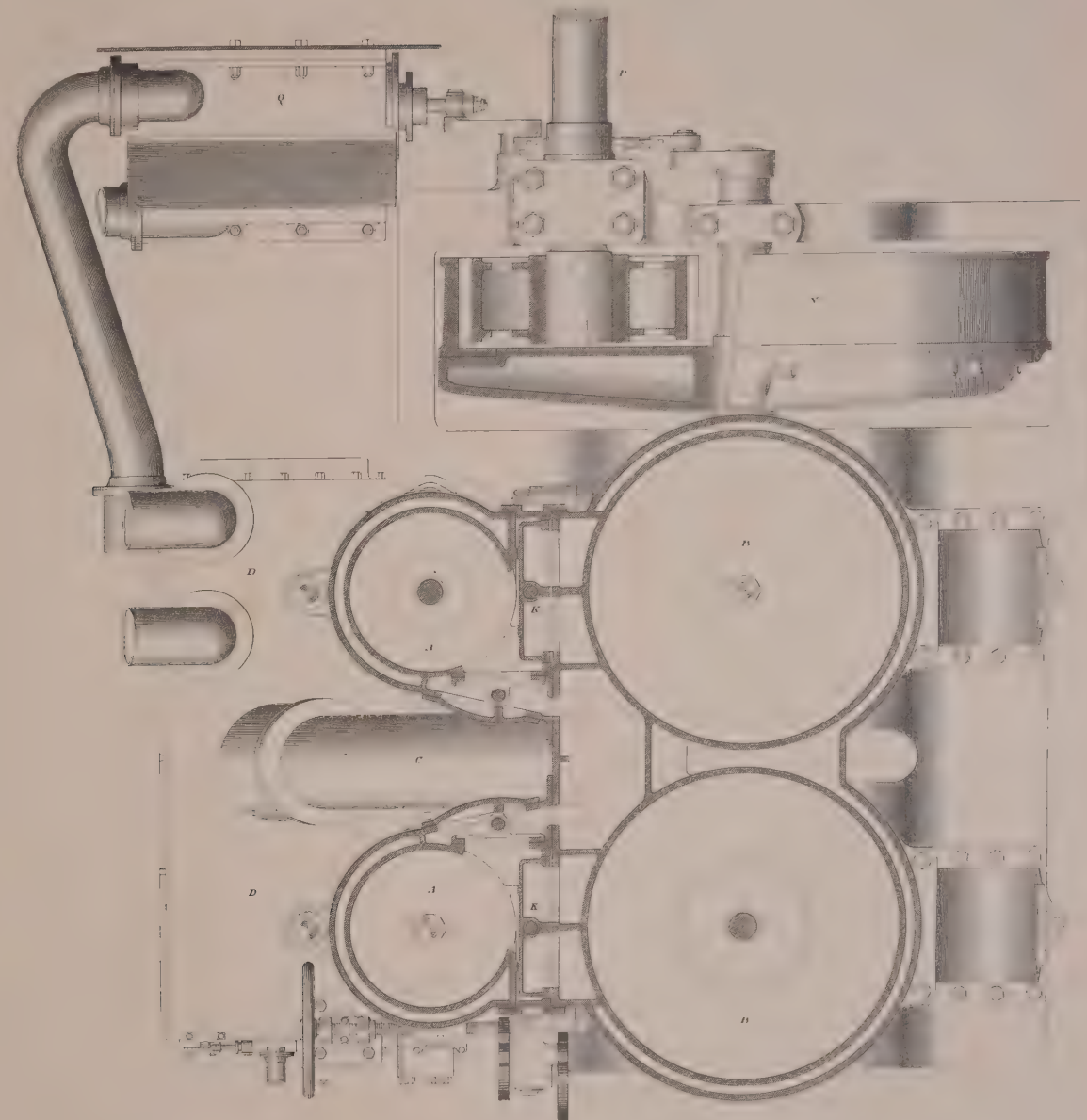
PAIR OF EXPANSIVE DOUBLE CYLINDER INVERTED GEARED ENGINES

OF 250 NOMINAL H.P. WITH SUPERHEATING AND SURFACE CONDENSATION
MADE FOR THE SCREW STEAM SHIP CALABAR OF 1200 TONS REGISTER
BY MESSRS RANDOLPH ELDER & COY ENGINEERS & SHIPBUILDERS, GLASGOW

THWARTSHIP SECTION THROUGH FORWARD ENGINE



HORIZONTAL SECTION THROUGH CYLINDERS

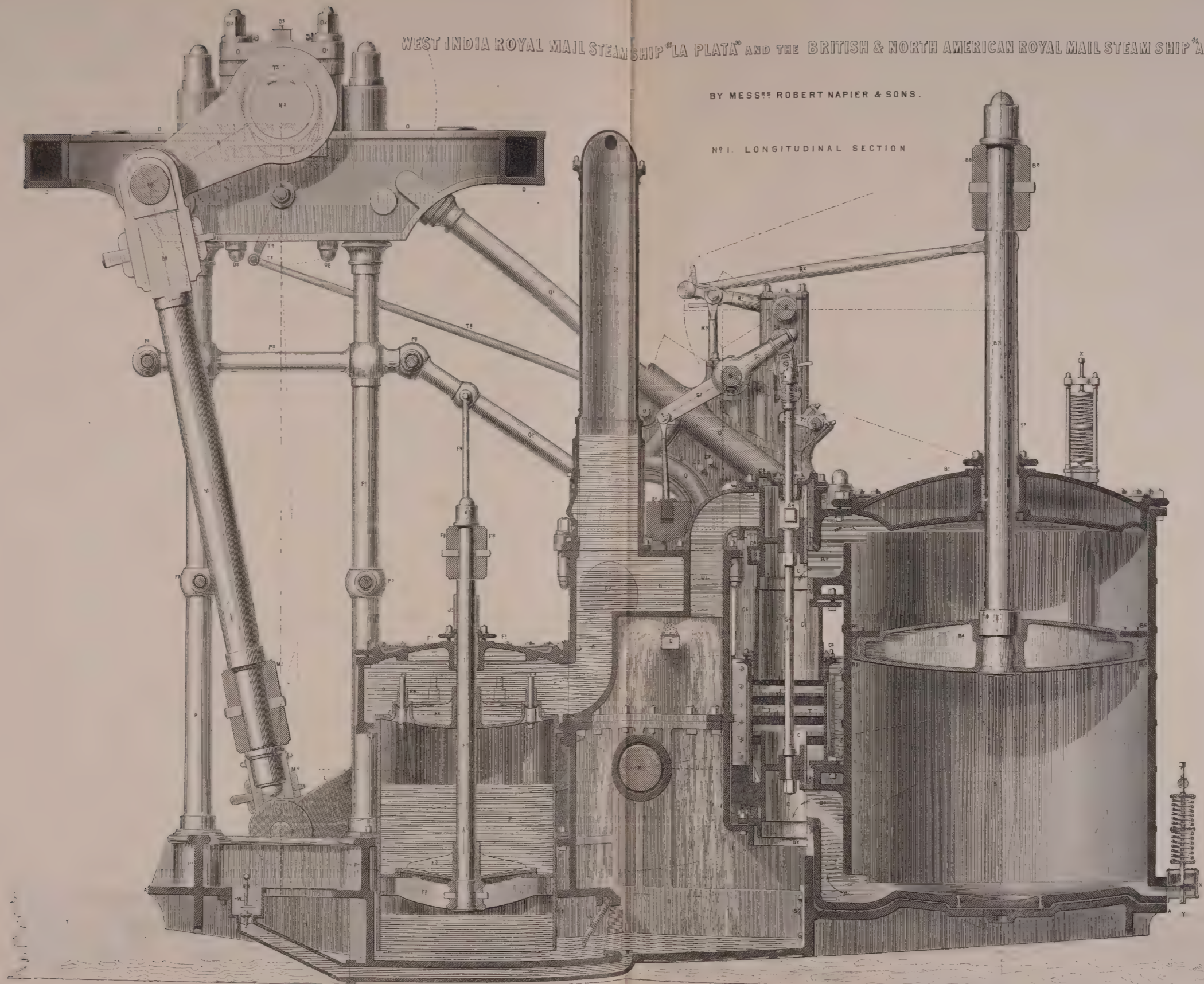


10 Feet

WEST INDIA ROYAL MAIL STEAMSHIP "LA PLATA" AND THE BRITISH & NORTH AMERICAN ROYAL MAIL STEAMSHIP "ARABIA"

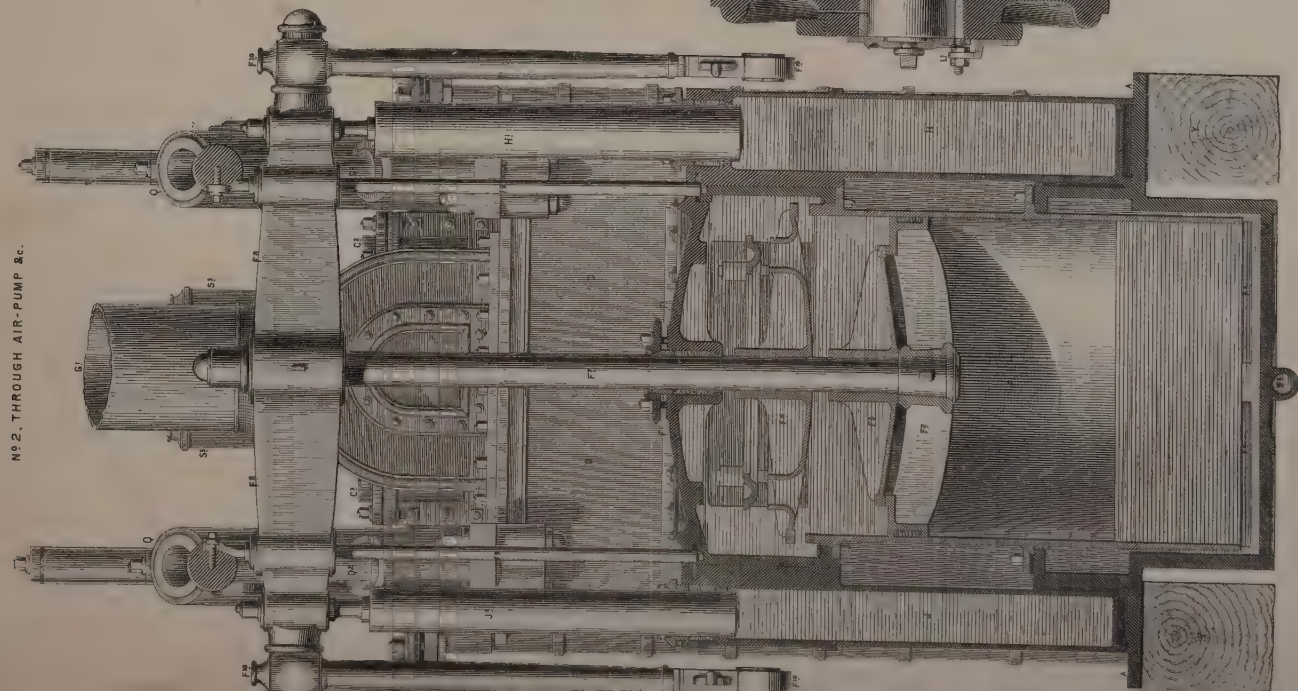
BY MESSRS ROBERT NAPIER & SONS.

Nº 1. LONGITUDINAL SECTION



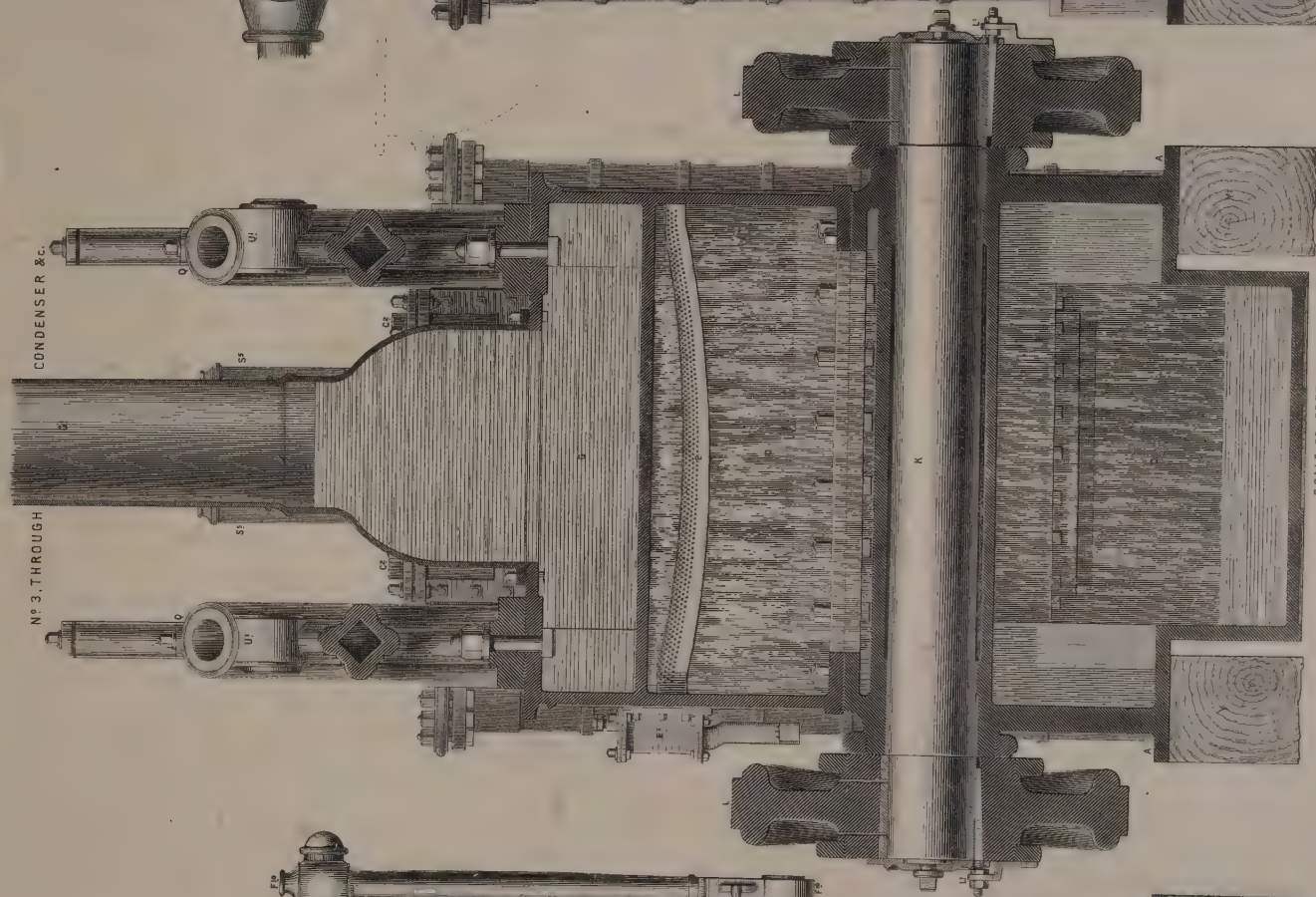
SCALE OF FEET

NO 2, THROUGH AIR-PUMP &c.

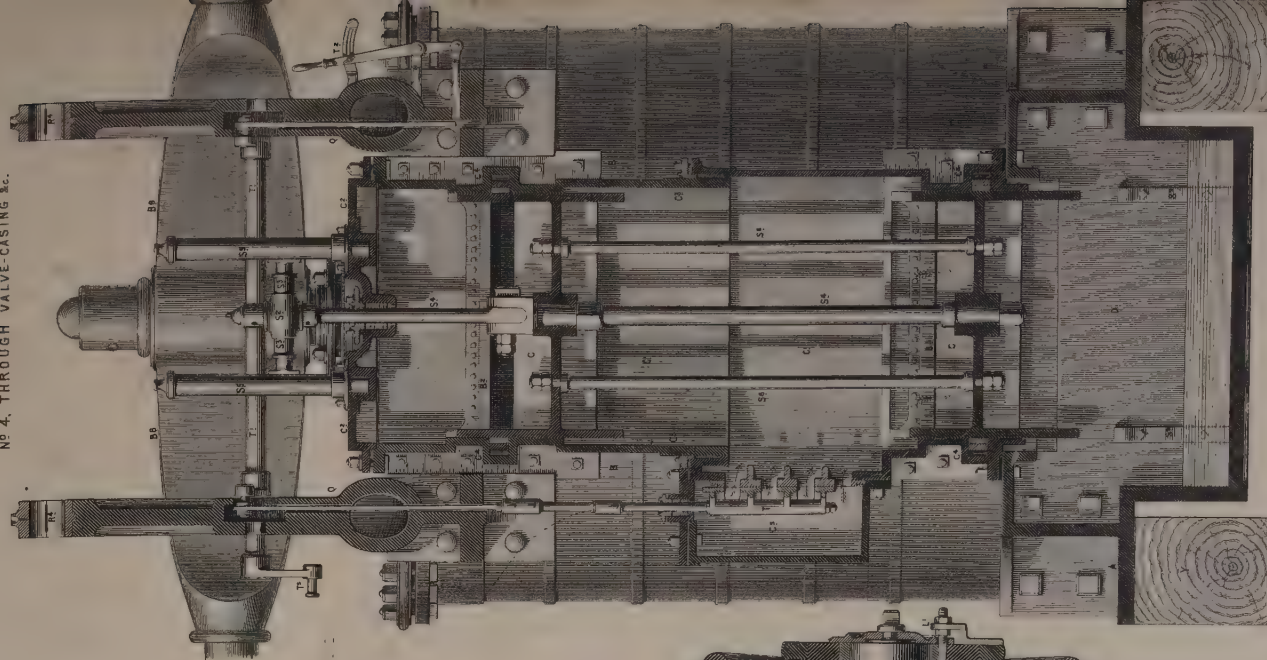


NO 3, THROUGH

CONDENSER &c.



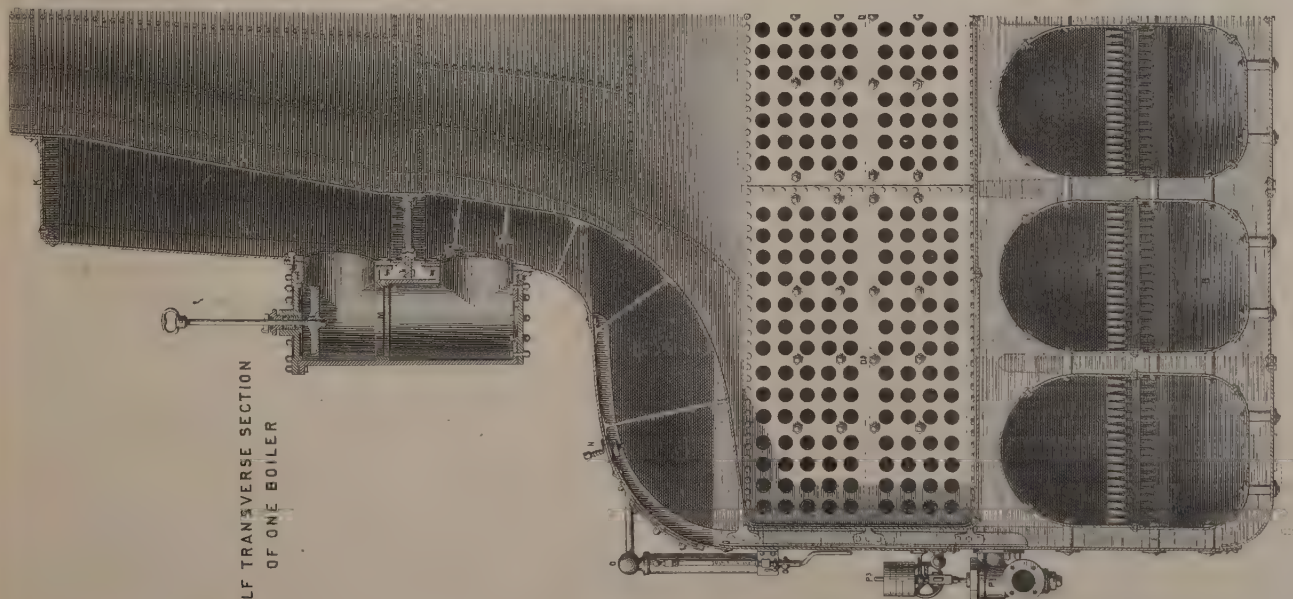
NO 4, THROUGH VALVE-CASING &c.



SCALE OF FEET



HALF TRANSVERSE SECTION
OF ONE BOILER



LONGITUDINAL SECTION
OF ONE BOILER

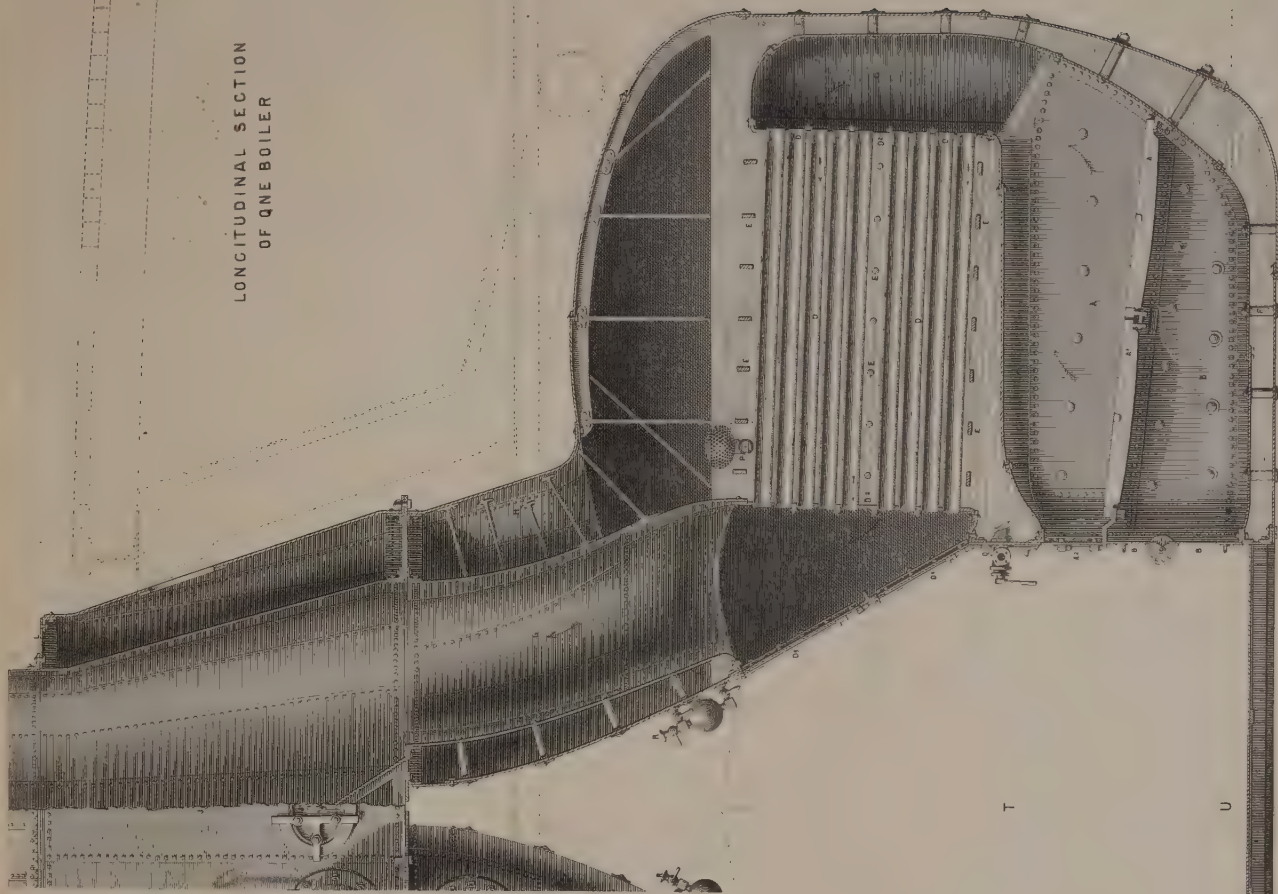


Fig. 1.

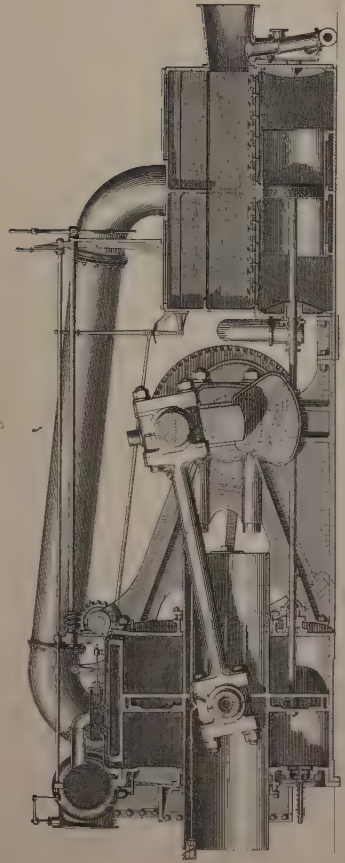


Fig. 3.

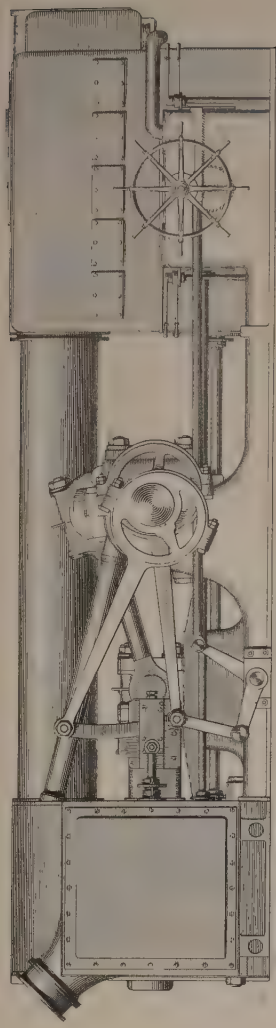


Fig. 2.

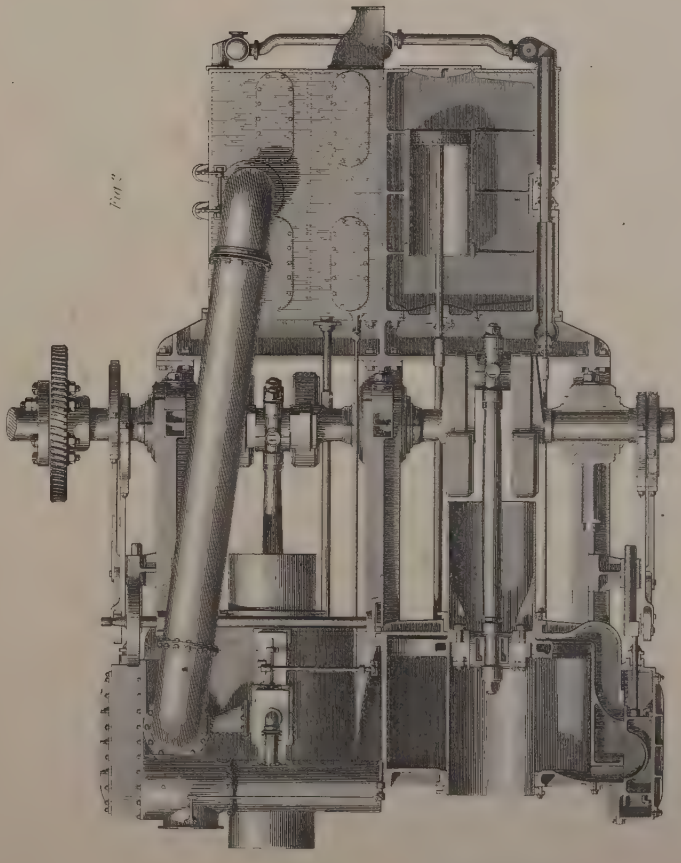
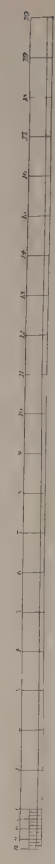
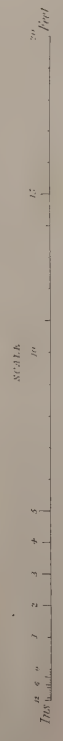
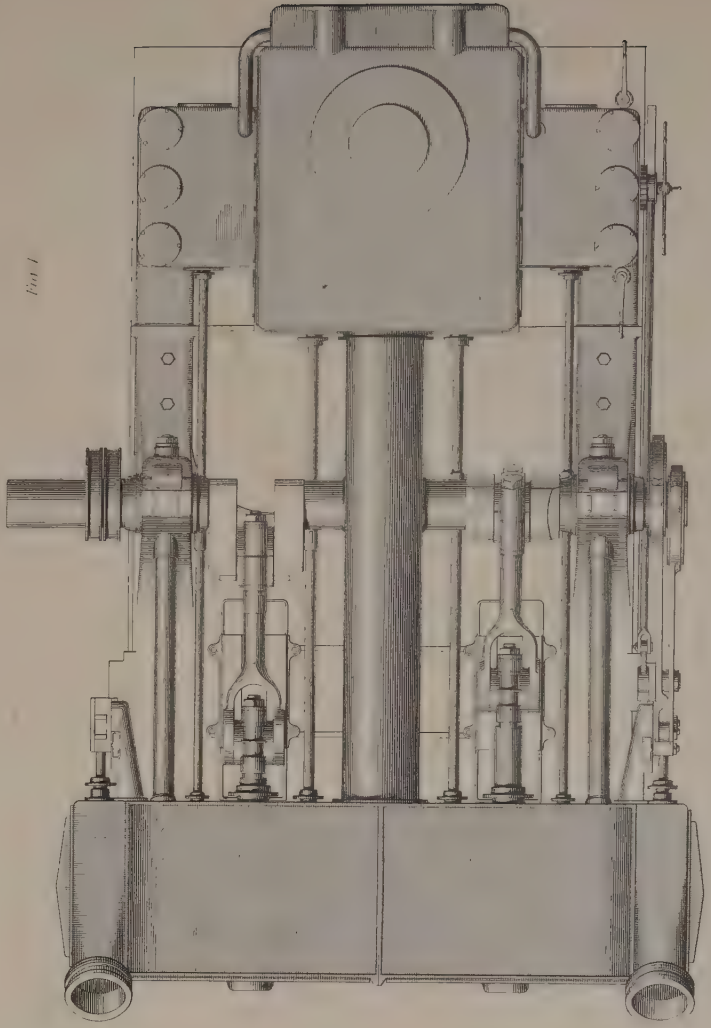
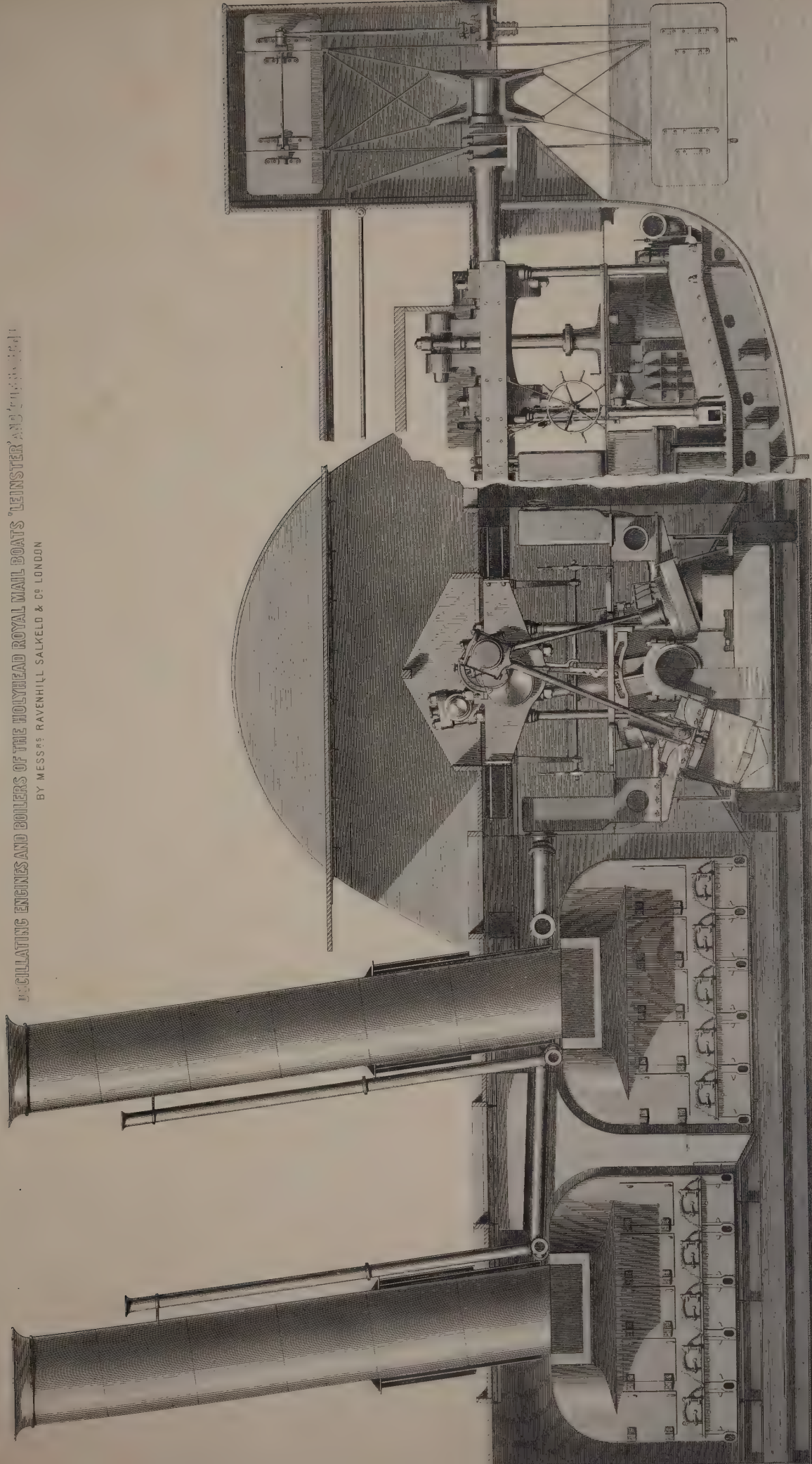


Fig. 4.

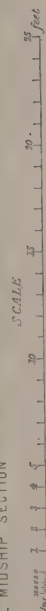


WILLIAM WACKENZIE, GLASGOW, EDINBURGH & LONDON

OSCILLATING ENGINES AND BOILERS OF THE HOLYHEAD ROYAL MAIL BOATS "LEINSTER" AND "MUNSTER"
BY MESSRS RAVENHILL SALKELD & CO LONDON



LONGITUDINAL MIDSHIP SECTION



TRANSVERSE SECTION

WILLIAM MACKENZIE, GLASGOW, EDINBURGH & LONDON

DESIGNED BY CHARLES F. HENWOOD ESQ. N.A. ACCORDING TO THE SYSTEM OF CAPTAIN COWPER P. COLES. R.N.

ELEVATION

Fig. 1.

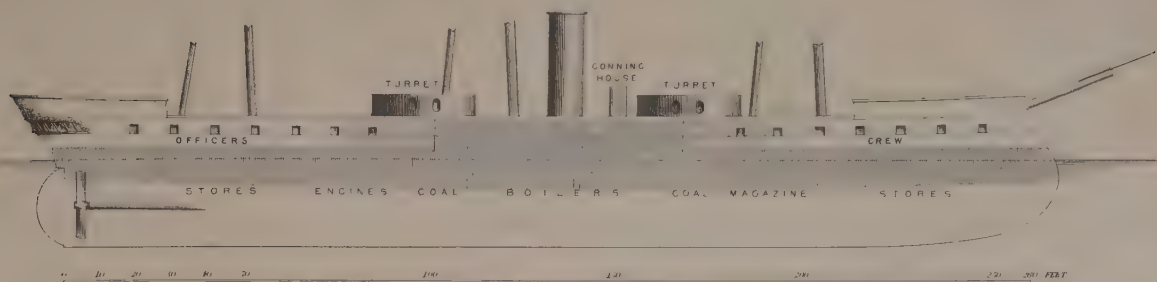


Fig. 2.

PLAN OF UPPER DECK



CROSS SECTION AT TURRET

Fig. 3.

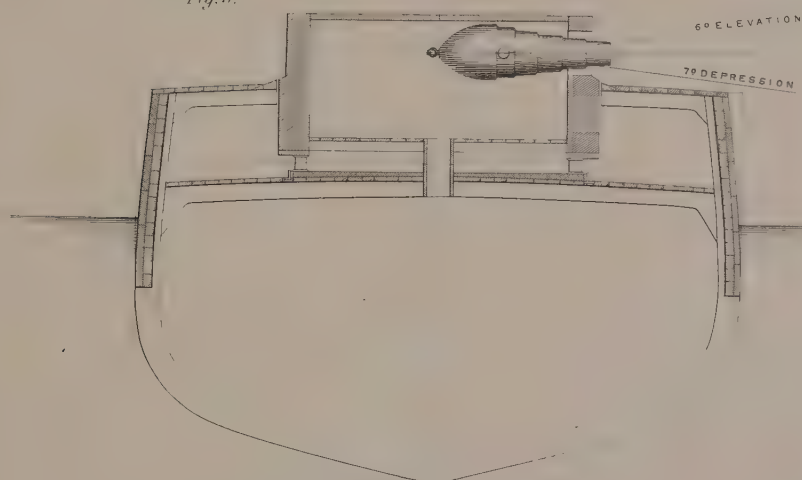
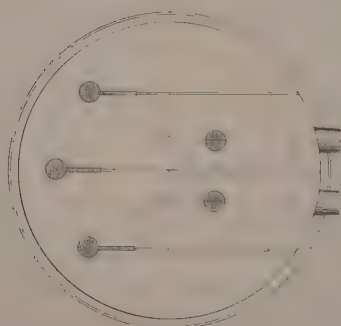


Fig. 4.

PLAN OF TURRET



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